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OF
THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

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The Society as a body is not responsible for the statements of facts or opinions advanced in papers or discussions. C55

THE JOURNAL

OF

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

VOL. 32 APRIL 1910 NUMBER 4

THE St. Louis monthly meeting for April will be held Saturday evening, April 9, with the American Institute of Electrical Engineers and The Engineers' Club of St. Louis coöperating. The subject of the meeting will be The Electric Drive in the Machine Shop, with four papers constituting a comprehensive symposium, both from the standpoint of the mechanical equipment of machine tools and from the economic side of the saving to be effected by the installation of different types of motor drives.

Three of the papers are contributed by The American Society of Mechanical Engineers as follows: The Economy of the Electric Drive, by A. L. DeLeeuw, Mem. Am. Soc. M.E., published in The Journal for November 1909; The Economical Features of Electric Motor Applications by Charles Robbins, Assoc.Mem.A.I.E.E.; Mechanical Features of Electric Driving, by John Riddell, Mem. Am.Soc.M.E. The two latter papers are published in this number of The Journal.

The American Institute of Electrical Engineers have contributed a paper by Charles Fair upon Selection and Methods of Application of Motors and Controllers.

MEETING IN NEW YORK APRIL 12

The New York monthly meeting for April will be held in the Auditorium of the Engineering Societies' Building, Tuesday evening, April 12, the American Institute of Electrical Engineers coöperating. The subject of the The Electric Drive in the Machine Shop will be

discussed and the four papers previously mentioned in the announcement of the April St. Louis meeting will be presented by Messrs. A. L. De Leeuw, Charles Robbins, John Riddell and Charles Fair.

There will be discussions by Messrs. Gano Dunn, Vice-president of the Crocker-Wheeler Company, Ampere, N. J., Clarence L. Collens, President, Reliance Electric and Engineering Company, Cleveland, H. A. Hornor, Electrical Engineer, Philadelphia, Charles Day of Dodge & Day, Philadelphia, Henry Hess of Philadelphia, and other prominent engineers, among them an official representative of the Machine Tool Builders Association, who will report upon the efforts of this Association at standardization of electric motor equipment.

MEETING IN BOSTON APRIL 27

The Boston monthly meeting of the Society, April 27, in which the American Institute of Electrical Engineers, Boston Section, and the Boston Society of Civil Engineers, will coöperate, will be the occasion of the presentation of a paper by Prof. C. M. Allen of Worcester Polytechnic Institute, Mem. Am. Soc. M. E., on The Testing of Water Wheels after Installation, published in this issue of The Journal. The meeting will be held at 8 p.m., in the hall of the Edison Electric Illuminating Company, 39 Boylston St.

Preliminary announcement is also made of the May meeting, to be held May 18, in charge of the Boston Section of the American Institute of Electrical Engineers, with The American Society of Mechanical Engineers and the Boston Society of Civil Engineers coöperating. The gathering will be addressed by Lewis B. Stilwell, President Am. Inst. E. E., on the subject of conservation, especially of water power. The place of meeting will be announced by the American Institute of Electrical Engineers to members of the three societies.

MEETING IN NEW YORK, MARCH 8

The New York monthly meeting for March, in which the American Institute of Electrical Engineers coöperated, was the largest monthly meeting of the Society held during the present season. The paper by H. G. Stott, Mem. Am. Soc. M. E., and J. S. Piggott, on tests on the 15,000-kw. steam-engine-turbine unit located in the 59th Street Station of the Interborough Rapid Transit Company of New York, aroused an unusual amount of interest.

The attendance at the meeting was nearly 700, and the audience remained until a late hour to listen to the comments of those who wished to speak from the floor.

After the presentation by Mr. Stott of the main facts regarding these tests and the installation of the low-pressure turbine on which the tests were made, Mr. Piggott gave an interesting account of the methods used in conducting the tests, illustrating his talk with lantern slides.

A summary of the discussion at this meeting, together with an abstract of that given at the recent Boston meeting which Mr. Stott addressed on the subject of Low-pressure Turbines, and the topical discussion on Steam Turbines given at a meeting of the Society in St. Louis, will be published together in a subsequent issue.

MEETING IN BOSTON, MARCH 11

A meeting of the Society was held in Boston, March 11, in which the Boston Section of the American Institute of Electrical Engineers and the Boston Society of Civil Engineers coöperated. The total attendance was 125. Chairman Hollis said in his introductory remarks that in future all these gatherings would probably be meetings not of one society but of engineers in general, thus inaugurating a new movement among the engineering societies of the country of pretty thorough coöperation. As a result of the meeting on January 21, a committee of fifteen, representative of twelve of the thirty or forty different societies in the vicinity of Boston having especial scientific and engineering interests, had been formed and had under consideration the possibility of erecting an engineering building and perhaps organizing within it a club, as well as providing rooms for the various societies and a meeting place for such gatherings as this. This movement, he thought, would not only bring together the various societies but should serve to relate the engineering profession more closely to civic questions in Boston.

Magnus W. Alexander of Lynn, Mass., Mem.Am.Soc.M.E., presented his paper on The Training of Men, a Necessary Part of a Modern Factory System, published in The Journal for January. This was discussed by Henry E. Rhoades, Past Assistant Engineer, U. S. N., Retired, Prof. Chas. F. Park and R. H. Smith of the Massachusetts Institute of Technology, Prof. Ira N. Hollis, Mem. Am.Soc.M.E., Prof. Gardner C. Anthony, Mem.Am.Soc.M.E., Prof. Peter Schwamb, Mem.Am.Soc.M.E., Luther D. Burlingame, Mem.Am.Soc.M.E., G.

C. Ewing, Assoc.Am.Inst.E.E., S. Fred Smith, President, New England Section of the National Electric Light Association, H. S. Knowlton, Prof. Edward F. Miller, of the Massachusetts Institute of Technology, Mem.Am.Soc.M.E., and Dickerson G. Baker, Mem.Am.Soc.M.E.

MEETING IN ST. LOUIS, MARCH 12

A topical discussion was held by the Society and the Engineers' Club of St. Louis, in the club rooms on Olive Street, Saturday evening, March 12, on the subject of Flywheels. The discussion was opened by Dr. C. H. Benjamin, Mem.Am.Soc.M.E., Dean of the Schools of Engineering of Purdue University, who reviewed the recent developments in testing flywheels and gave an account of the latest apparatus and testing pit for this purpose at Purdue University. There were present at the meeting, Jesse M. Smith, Past-President, Col. E. D. Meier, Vice-President, and Calvin W. Rice, Secretary of this Society. An informal dinner was held before the meeting at the Mercantile Club, attended by the guests and local members.

As is customary when it is impossible for the president to be present, a vice-president is invited to preside, and Col. E. D. Meier, Vice-President, acted as chairman. The presiding officer and William H. Bryan, Chairman of the Local Committee, took the opportunity, previous to the topical discussion, to give the greetings of the Council to the members, and to explain the interest on the part of the Council in the meetings of the Society in the several cities and the appreciation by the Local Committee of their efforts. It was explained that all meetings of the Society are conducted under the same rules, wherever held, the national character of the Society being in this way developed; and at the same time the members are enabled to get the greatest benefits from their association.

Professor Benjamin, who has for many years conducted experiments on flywheels with a view to determining the bursting strength of different types of wheels, followed with his remarks on the subject, after which there was a general discussion.

SPRING MEETING ATLANTIC CITY

RAILROAD TRANSPORTATION NOTICE

Arrangements for hotel, transportation and Pullman car accommodations should be made personally.

Special concessions have been secured for members and guests attending the Spring Meeting in Atlantic City, May 31 to June 3, 1910.

The special rate of a fare and three-fifths for the round trip, on the certificate plan, is granted when the regular fare is 75 cents and upwards, from territory specified below. Read item *g*.

a Buy your ticket at full fare for the going journey, between May 27 and June 2 inclusive. At the same time request a certificate, *not a receipt*. This ticket and certificate should be secured at least half an hour before the departure of the train.

b Certificates are not kept at all stations. Ask your station agent whether he has certificates and through tickets. If not, he will tell you the nearest station where they can be obtained. Buy a local ticket to that point, and there get your certificate and through ticket.

c On arrival at the meeting, present your certificate to S. Edgar Whitaker, office manager at the Headquarters. A fee of 25 cents will be collected for each certificate validated. No certificate can be validated after June 3.

d An agent of the Trunk Line Association will validate certificates, June 1, 2, 3. No refund of fare will be made on account of failure to have certificate validated.

e One-hundred certificates and round trip tickets must be presented for validation before the plan is operative. This makes it important to show the return portion of your round trip ticket at Headquarters.

f If certificate is validated, a return ticket to destination can be purchased, up to June 8, on the same route over which the purchaser came, at three-fifths the rate.

g Members and guests from New York City should buy the regular round trip tickets at $5 and show the return portion to Mr. Whitaker or the validating agent.

The special rate is granted only for the following:

The Trunk Line Association:

All of New York east of a line running from Buffalo to Salamanca, all of Pennsylvania east of the Ohio River, all of New Jersey, Delaware and Maryland; also that portion of West Virginia and Virginia north of a line running through Huntington, Charleston, White Sulphur Springs, Charlottesville, and Washington, D. C.

The Central Passenger Association:

The portion of Illinois south of a line from Chicago through Peoria to Keokuk and east of the Mississippi River, the States of Indiana, and Ohio, the portion of Pennsylvania and New York north and west of the Ohio River, Salamanca and Buffalo, and that portion of Michigan between Lakes Michigan and Huron.

The New England Passenger Association except via N. Y. O. & W. R. R. and Eastern Steamship Co. The Rutland R. R. participates in fares reading via its road:

Maine, New Hampshire, Vermont, Massachusetts, Rhode Island and Connecticut.

The Western Passenger Association offers revised one-way fares, on the basis of two cents per mile, to Chicago, Peoria and St. Louis; these three places are points in the Central Passenger Association, and from these points purchase round trip tickets, in the manner outlined in the preceding paragraphs:

North Dakota, South Dakota, Nebraska, Kansas, Colorado, east of a north and south line through Denver, Iowa, Minnesota, Wisconsin, Missouri, north of a line through Kansas, Jefferson City and St. Louis; Illinois, north of a line from Chicago through Peoria to Keokuk.

The Eastern Canadian Passenger Association:

Canadian territory east of and including Port Arthur; Sault Ste. Marie, Sarnia and Windsor, Ont.

MEETING IN GREAT BRITAIN

Invitations have been received through the Verein Deutscher Ingenieure, to the Königliche Technische Hochschule of Berlin and of Danzig-Langfuhr, as well as the following industrial works:

INDUSTRIELLE WERKE

Accumulatorenfabrik A. G., Hagen i/Westif.
Düsseldorfer Eisenbahnbedarf vorm. Carl Weyer & Co., Düsseldorf Oberbilk-Wegmann & Co., Kassel
Van der Zypen & Charlier, G.m.b.H., Cöln-Deutz
J. Goossens, Aachen (Eschweiler-Aue)
Eschweiler Bergwerksverein, Eschweiler
Gebr. Stumm, G.m.b.H., Neunkirchen (Bez. Trier)
Rheinisch-Nassauische Bergwerks- & Hütten Akt. Ges., Stolberg i/Rhld.
Ehrhardt & Sehmer, G.m.b.H., Schleifmühle, Post Saarbrücken
Dingler'sche Maschinenfabrik A. G., Zweibrücken (Pfalz)
Röchling'sche Eisen- und Stahlwerke, G.m.b.H., Völklingen (Saar)
Maschinen- und Armaturenfabrik vorm. Klein, Schanzlin & Becker, Frankenthal
Stahlwerke Mannheim, Rheinau bei Mannheim
Badische Aktiengesellschaft für Rheinschiffahrt und Seetransport, Mannheim
Heinrich Lanz, Lokomobilen und landwirtschaftliche Maschinen Mannheim (Firma bittet um namentliches Verzeichnis der Teilnehmer nebst Angabe der Firma und deren Branche. Ingenieure solcher Fabriken, die sich mit dem Bau von Lokomobilen und landwirtschaftlichen Maschinen beschäftigen, können an der Beschtigung nicht teilnehmen.)
Aktiengesellschaft vorm. Burgeff & Co., Schaumweinfabrik, Hochheim a/M.
L. Schuler, Werkzeugmaschinenfabrik, Göppingen
Aktiengesellschaft L. A. Riedinger, Maschinen-& Broncewarenfabrik, Augsburg
Maschinenfabrik Augsburg-Nürnberg, Aktiengesellschaft, Nürnberg
Siemens-Schuckertwerke, Nürnberg
Werkzeugmaschinenfabrik Union, vorm. Diehl, Chemnitz
J. E. Reinecker, Werkzeugmaschinenfabrik, Chemnitz-Gablenz
Dresdner Bohrmaschinenfabrik A. G. vorm. Bernhard Fischer & Winsch Dresden-A., Zwickauerstrasse 41–45
Aktiengesellschaft vorm. Gebr. Seck. Mühlenbauanstalt, Dresden-A.
Telefonfabrik Aktiengesellschaft vorm. J. Berliner, Hannover
Beuchelt & Co., Grünberg i/Schles.
A. Borsig, Tegel-Berlin
Siemens-Schuckertwerke, Berlin

Siemens & Halske, Aktiengesellschaft, Berlin
H. Aron, Elektrizitätszahlerfabrik, G.m.b.H., Berlin-Charlottenburg
Allgemeine Elektrizitätsgesellschaft, Berlin
Ludwig Loewe & Co., Aktiengesellschaft Berlin, NW Huttenstr. 17-20
R. Wolf, Lokomobilen, Magdeburg-Buckau
Deutsche Wagenbau-und Leihgesellschaft m.b.H., Danzig
Aktiengesellschaft Amme, Giesecke & Konegen, Mühlenbaustalt Braunschweig
Breslauer Aktiengesellschaft für Eisenbahn-Wagenbau, Breslau

WISSENSCHAFTLICHE INSTITUTE

Physikalisch-Tehnische Reichsanstalt, Charlottenburg-Berlin
Maschinenlaboratorium der Kgl. Techn. Hochschule, Berlin
Maschinentechnisches Laboratorium der Kgl. Techn. Hochschule Danzig
Maschinenlaboratorium der Kgl. Techn. Hochschule Aachen
Material prufüngsanstalt, Laboratorium und Institut der Grossherz. Techn. Hochschule Darmstadt
Materialprüfungsanstalt und Ingenierlaboratorium der Kgl. Techn. Hochschule Stuttgart
Grossherz. Badische chemisch-technische Prüfungs-und Versuchsanstalt, Lehr- und Versuchs-Gasanstalt und andere Institute an der Technischen Hochschule Karlsruhe
Königliches Material prüfungsamt Gross-Lichterfelde West. (Berlin)
Maschinenlaboratorium und elektrotechnisches Laboratorium der Kgl. Technischen Hochschule Breslau. (z. Zt. noch im Ausbau; da die Laboratorien im Juli ds. Jrs. vielleicht betriebsfähig sein können, empfiehlt sich eine vorherige Anfrage beim Rektorat der Hochschule.)

Translations of the invitations from the Königliche Technische Hochschule of Berlin and of Danzig-Langfuhr, with the accompanying letter from the Verein deutscher Ingenieure, follow.

Members of the Society desiring to avail themselves of these courteous invitations will correspond with the Secretary that he may secure them letters of introduction.

VEREIN DEUTSCHER INGENIEURE
Berlin, N. W., Charlottenstrasse, 43

Col. E. D. Meier
Care of Heine Safety Boiler Co.,
11 Broadway, New York.

Berlin, January 31, 1910

Referring to your correspondence with Dr. Ing. Diesel of Munich, concerning the inspection of German industrial works and scientific institutions by members of The American Society of Mechanical Engineers, we have the honor to send you a list of firms and institutions which have declared themselves ready to receive these gentlemen, and enclose also several letters.

As we are not informed of the traveling route of the members of your highly appreciated Society, and do not know their wishes or preferences, we consider

it advisable that The American Society of Mechanical Engineers should correspond directly with such works and institutions as they would like to visit and arrange the necessary details.

With highest regards,

Business office of the
VEREIN DEUTSCHER INGENIEURE
Linde

MASCHINENBAULABORATORIUM DER KONIGL. TECHNISCHEN HOCHSCHULE ZU BERLIN

Vorsteher: Prof. E. Josse

Charlottenburg, November, 15, 1909.

Highly honored gentlemen:

I have been informed that on the occasion of your visit to England and the British Institution of Mechanical Engineers, you also think of visiting Germany in order to examine industrial works, scientific institutions, etc.

We would greet with pleasure your visit to the Königliche Technische Hochschule of Charlottenburg, and we would then gladly take the opportunity to show you the various laboratories of the Hochschule, especially the Mechanical Laboratory in all its details.

I beg you therefore to embody in your programme a visit to the above named institution and sign, with especial regards

Cordially,
JOSSE
Professor.

To The American Society of Mechanical Engineers.

MACHINENTECHNISCHES LABORATORIUM DER KGL. TECHNISCHE HOCHSCHULE
Prof. A. Wagener

Danzig-Langfuhr, November, 15, 1909.

To the American Society of Mechanical Engineers:

Having learned through the Verein deutscher Ingenieure that at the conclusion of the Annual Meeting of The American Society of Mechanical Engineers which is to take place about the end of July, 1910, in London, a visit to German industrial works and scientific institutions, is planned, I have the honor to extend a cordial invitation to The American Society of Mechanical Engineers to visit the mechanical laboratory of the Königliche Technische Hochschule at Danzig.

With highest regards

Respectfully
A. WAGENER.

SOCIETY AFFAIRS

MEETING OF THE COUNCIL

A regular meeting of the Council was held March 8, 1910, with E. D. Meier, Vice-President, presiding. There were present Charles Whiting Baker, George M. Bond, John R. Freeman, H. L. Gantt, James Hartness, Alex. C. Humphreys, F. R. Hutton, E. D. Meier, I. E. Moultrop, H. G. Reist, Jesse M. Smith, Arthur M. Waitt, and the Secretary. Regrets were received from President Westinghouse and W. F. M. Goss, Vice-President.

The minutes of the meeting of February 8 were read and approved, after being amended to read,

Voted: That the proposed amendments with the report of the Finance Committee, be referred to the Committee on Constitution and By-Laws jointly with the Finance Committee for report back to the Council.

The Secretary announced the deaths of A. M. Goodale and Percy A. Sanguinetti.

The resignations of A. E. Holcomb and H. J. Scales were accepted with regret. The membership of W. W. Bigelow was declared to have lapsed.

The Secretary read the invitation of President Aspinwall of the Institution of Mechanical Engineers, to members of the Council and Past Presidents of this Society, to a dinner on the evening of Monday, July 25. A total of 208 are planning to attend the Joint Meeting in England in July, of whom 135 have arranged to sail on the Celtic.

Letters were read from the Königliche Technische Hochschule of Berlin and of Danzig-Langfuhr, and the Verein deutscher Ingenieure extending invitations to the visiting members of the Society.

Voted: To appoint the following Committee on Land and Building Fund: H. F. Holloway, I. E. Moultrop, F. H. Stillman, Morris L. Cooke, George A. Orrok.

The Secretary stated that the memorial window to Sir Benjamin Baker, Hon. Mem.Am.Soc.M.E., deceased, was presented by the Earl of Cromer in behalf of the subscribers and accepted by the Dean of Westminster Abbey in December 1909, and that report including

a colored illustration of the window was in preparation by the Institution of Civil Engineers.

Resolutions were read from the members of the Society resident in St. Louis, together with the action of the Executive Committee, as follows:

Voted: That a committee to consist of Jesse M. Smith, Past-President, E. D. Meier, Vice-President, I. E. Moultrop, Manager, Willis E. Hall, Chairman of Meetings Committee, and Calvin W. Rice, Secretary, be appointed to represent the Council at the meeting of the Society in St. Louis, March 12.

The committee is instructed to present the greetings of the Council to the St. Louis members, to gather information regarding the meetings of the Society in St. Louis, and to report its findings to the Council together with its recommendations, as soon as practicable.

The communication of February 15 signed by Mr. Wm. H. Bryan, from the members of the Society in St. Louis, together with the communication of February 25 with respect to same from the Meetings Committee, was read; after consideration the Chairman of the Meetings Committee and the Secretary were requested to draft a letter for submission to the Executive Committee and the Council, which shall represent the views of the latter body.

The communication of February 23 from Mr. Wm. H. Bryan, representing the members of the Society in St. Louis, requesting permission to hold a meeting of the Society on March 12, referred by the Meetings Committee to the Executive Committee, was referred back to the Meetings Committee with power, with the suggestion that authorization be given for a topical discussion, namely, The Recent Developments in the Bursting of Flywheels and Pulleys, by Prof. C. H. Benjamin of Purdue University, the authorization to be given under By-Law 23.

Notice was given of the purpose of the Committee on Constitution and By-Laws to amend By-Laws 11, 16, 17, 18, as follows:

FEES AND DUES

B 16 The initiation fee and the annual dues for the first year shall be due and payable on the first day of the month following the date of the election of a Member, Associate or Junior. The annual dues for each ensuing year shall be due and payable in advance on the corresponding day in each year thereafter. Upon the payment of the initiation fee and the annual dues for the first year, the person elected shall be entitled to the rights and privileges of membership in the grade to which he was elected. The date of payment of a member's annual dues may be changed to the first day of any other month, and a *pro-rata* adjustment of the dues made, by application to the secretary.

B 17 A Member, Associate or Junior in arrears for dues for one year, on the first day of October previous to the Annual Meeting, shall not be entitled to

vote, or to receive the Transactions or the publications issued by the Society thereafter until such dues have been paid. Should the arrears for dues or otherwise be for more than two years, the name of such person shall be presented to the Council for such action as it deems advisable under C 24. Should the right to vote, or to receive the publications of the Society be questioned, the books of the Society shall be conclusive evidence.

B 18 The Council may, in its discretion, restore to membership any person dropped from the rolls for non-payment of dues, or otherwise, upon such terms and conditions as it may at the time deem best for the interests of the Society.

ELECTION OF MEMBERS

B 11 Each person elected to membership, except an Honorary Member must subscribe to the Constitution, By-Laws, and Rules of the Society, and pay the initiation fee before he can receive a certificate of membership in the Society. Resignations from membership shall be presented to the Council for action.

It was also voted to approve the adoption of B 22 and B 24, as published in another part of this issue of The Journal.

Voted: To amend Rule 24 to read as follows:

R 24 Engineers and others not members of the American Society, but desiring to participate in the meeting of the Section, may enroll themselves as affiliates as heretofore provided with the approval of the Executive of the section. Such affiliates shall have the privilege of presenting papers and taking part in the discussions. They shall pay $5 per annum which shall be due and payable in advance, on October 1 of each year of their enrollment, and shall thereby be entitled to receive the regular issues of The Journal for a period covered by their dues.

Voted: To amend Rule 29 to read:

R 29 The American Society of Mechanical Engineers will furnish monthly issues of The Journal to all members of affiliated organizations who are not members of The American Society of Mechanical Engineers upon the payment by each of two dollars per year, such payment being due January 1 of each year. The American Society of Mechanical Engineers will furnish gratis to each affiliated body, extra copies of advance papers for use at its meetings, the number furnished to be agreed upon at the discretion of the Secretary.

Voted: To approve the report of the Committee on International Standard for Pipe Threads.

Voted: To refer the communication of John Riddell, suggesting the preservation of models of epoch-making inventions that would be of interest in showing the development of engineering, to a committee of five, consisting of Charles Wallace Hunt, Col. E. A. Stevens, John Riddell, F. R. Hutton, Edward Van Winkle, to report what action if any, should in their judgment be taken. The letter is appended to this report.

Voted: To approve the recommendation of the Executive Committee, with regard to Col. R. S. Crompton's communication respecting the desirability of an international standard for machine screws; that the Secretary be directed to reply that inasmuch as the Society is to meet in England this summer the Institution would possibly be pleased to invite the Society to a conference on this subject; if so, our Society would be pleased to respond by appointing a committee to confer with a committee of the Institution. It was also advised that a suggestion be included regarding the coöperation of the Automobile Engineers of America.

A communication was read from the Verein Deutscher Ingenieure, inviting the coöperation of the Society in securing biographies and reminiscences of our engineers, and it was voted to refer this to the Committee on Society History.

A letter was read from L. B. Stillwell, President of the American Institute of Electrical Engineers, expressing cordial coöperation in the meeting of the Society on April 12.

The meeting adjourned to April 12.

Amendments to B 22 and B 24 Approved by Council

The Council voted to approve under the provision of C 59 the adoption of B 22 and B 24. The Amendment is to take effect immediately, as provided.

FINANCE COMMITTEE

B 22 The Finance Committee shall consist of five Members or Associates. The term of office of one Member of the Committee shall expire at the end of each Annual Meeting. This Committee shall under the direction of the Council, have a supervision of the financial affairs of the society, including the books of account. The Committee may cause the accounts of the Society to be audited and approved annually by a chartered or other competent accountant. The Committee shall hold monthly meetings for the audit of bills and such other business as shall come before it and shall deliver to the Secretary for representation to the Council at the end of each fiscal year, a report of the financial condition of the Society for the past year, and also shall present therewith a detailed estimate of the probable income and expenditure of the Society for the following twelve months. It shall make recommendations to the Council as to investments, and when called upon by the Council, advise upon financial questions. It shall have charge of the making of all contracts and other obligations to pay money in the Society's work and the ordering of all expenditures thereunder.

PUBLICATION COMMITTEE

B 24 The Publication Committee shall consist of five Members or Associates. The term of office of one Member shall expire at the end of each Annual Meeting. The Committee shall review all papers and discussions which have been presented at the meetings, and shall decide what papers and discussions, or parts of the same, shall be printed in the Transactions of the Society. The Committes shall have the supervision of the monthly publication of the Society known ae "The Journal." The Committee will be expected to publish all such data as will be of assistance to engineers or investigators in their work. At the end of each fiscal year the Committee shall deliver to the Secretary for presentation to the Council a detailed report of its work.

Proposed Museum of Mechanic Arts

The communication previously referred to, by John Riddell, proposing a museum of Mechanic Arts, is as follows:

February 26, 1910

To the President and Council
The American Society of Mechanical Engineers,
29 West 39th Street, New York City.

Gentlemen:

The rapid advancement in Mechanic Arts in this country causes machinery and mechanical devices to be quickly superseded by newer developments. Such superseded devices are often most interesting, in studying the development of engineering, but unfortunately such devices disappear, leaving no record of what has been done. Think for a moment how interesting it would be to have the original and a few intervening forms of such American devices as the cotton gin, and a few of the numerous machines which we have made for harvesting cotton, the sewing machine, telephone, phonograph, linotype, telegraph, aeroplane, and other machines, while not of American origin, but to which we have greatly contributed in their development:—as the steam engine, plow, machine tools, general automatic machinery, bicycle, automobile, steam turbine, electric dynamo, and steel mill machinery.

In the South Kensington Museum the epoch-making inventions of Watt and Stephenson are carefully preserved, and visited by thousands of people from all civilized countries.

American Engineers have contributed so much to this work that it is appropriate that a record of the most important developments should be kept. I have thought for some time that a movement should be put on foot to establish a museum of mechanic arts, and that it is most appropriate that this should be done by The American Society of Mechanical Engineers.

It is important if anything is to be done in this line that action should not be delayed, since it will be readily understood that much material is available for such a museum now that could not be had ten years hence.

I am prompted to make this suggestion in view of the forthcoming trip abroad by many of your members and I might also suggest that some investigations ad report be made on the feasibility of such a scheme.

Yours truly

(*Signed*) JOHN RIDDELL.

Members who think a museum of such a character would be desirable are urged to make suggestions in regard to its scope as well as of articles that should be so preserved, or to manifest their interest by assisting to endow the museum or in any other manner.

STUDENT BRANCHES

BROOKLYN POLYTECHNIC INSTITUTE

William Kent, editor of Industrial Engineering and member of the Society, will give an address on Engineering and Common Sense, at the regular meeting of the Brooklyn Polytechnic Student Section, on Saturday evening, April 9, 1910, in the institute chapel, 85 Livingston St., Brooklyn.

PURDUE UNIVERSITY

On March 3, 1910, the Purdue Student Section was addressed by G. A. Weschler (1910), on The Manufacture of Cartridge Cases for the United States Navy. Mr. Weschler's eight years' experience in the Washington navy yard, two years of which were spent in the cartridge case shop, has made him familiar with a manufacturing process not generally known. The various steps in the manufacture were taken up in detail, the talk being illustrated by lantern slides.

STEVENS INSTITUTE OF TECHNOLOGY

The following lectures, of the schedule for 1909-1910, will be delivered before the Stevens Engineering Society: April 5, The Theory of Gyrostatic Motion, by Lewis A. Martin; April 12, Methods and System in Relation to Handling Concrete Work, by Frank B. Gilbreth, Mem.Am.Soc.M.E.; April 19, Notable Examples in Modern Construction, by John C. Ostrup; April 26, Development of the NewNavy, by David Watson Taylor; May 10, The Contribution of Photography to our Knowledge of the Stellar Universe, John A. Brashear, Hon. Mem. Am. Soc.M. E.

MASSACHUSETTS INSTITUTE OF TECHNOLOGY

The speakers at the annual meeting and banquet of the Mechanical Engineering Society of the Massachusetts Institute of Technology, were Prof. Gaetano Lanza, Mem.Am.Soc.M.E., Honorary Chairman of the organization, Colonel Locke, and C. C. Pierce. The following officers were elected for the ensuing year: Morril MacKenzie, Chairman; H. C. Brown, Vice-Chairman; Foster Russell, Secretary; H. S. Lord, Treasurer; D. P. Allen, H. S. Smith and A. F. Kenrick, Governing Committee.

NECROLOGY

JAMES HENRY BLESSING

James Henry Blessing was born in the village of French's Mills in Albany County, New York, September 14, 1837. At his father's death in 1849 he left school and in 1853 was apprenticed to the machinist trade for four years, with the firm of F. & T. Townsend. He remained with this firm until 1861, when he entered the U. S. Navy as acting assistant engineer. After the war he became engineer in charge of steam machinery of the Brooklyn Horse R. R. Co., returning to Albany in 1868 to act as superintendent of the foundry and machine works of Townsend & Jackson, successors to F. & T. Townsend.

In 1870 Mr. Blessing invented the return steam trap, the best known of his one hundred and twenty inventions. In July 1872 he left the employ of Townsend & Jackson to engage with Genl. Frederick Townsend in the manufacture and sale of these and other steam specialties, under the firm name of Townsend & Blessing. In June 1873 this firm sold their interest to the Albany Steam Trap Company and Mr. Blessing became secretary and treasurer and general superintendent of the company, and afterwards president.

Mr. Blessing was elected mayor of Albany in 1899. He was a member of the Society of Engineers of Eastern New York and of the Albany Historical and Art Society, and entered this Society in 1891. He died in Albany, February 21, 1910.

ALFRED MONTGOMERY GOODALE

Alfred Montgomery Goodale, Manager of the Society, 1898-1901, and a life member, died in Waltham, Mass., December 17, 1909.

He was born in Saco, Me., December 20, 1855, and educated in the public schools of that State, receiving from the Maine State College the degree of B.S. in 1875. He served for five years in the works of the Saco Water Power Machine Shop, Biddeford, Me., and the Bates Mills, Lewiston, Me., building, setting up and running cotton machinery; and in 1880 became superintendent of the Newton Mills, at New-

ton Upper Falls, Mass., afterwards acting as agent, from 1881 to 1883, and from 1884 to 1894, for the Hamilton Woolen Company of Amesbury, Mass., and the Boston Mfg. Co., of Waltham, Mass. Since 1894 he had been treasurer and a director of the Boston Mfg. Co. With each of these companies, Mr. Goodale had charge of the erection of new machinery, engines and boilers, and of the reorganization and improvements of the existing plants. In 1901 he started the firm of A. M. Goodale & Co., of Boston, brokers in cloths and yarns.

Mr. Goodale served on various city commissions and was a trustee of the Waltham hospital. He was a director of the New England and Northwestern Investment Company and of the Westfield Creel Company. Besides his club and Masonic connections he was president of the New England Cotton Manufacturers Association, and a member of this Society since 1886.

J. HENRY SIRICH, JR.

J. Henry Sirich, Jr., associate member of the Society, died January 22, 1910, at Bethlehem, Pa. He was born at Baltimore, Md., April 9, 1881, and received his technical education at the Baltimore Polytechnic Institute, from which he was graduated in the class of 1898. He served his apprenticeship at and afterwards entered the drafting room of the engineering works of Robert Poole & Son Company, now the Poole Engineering & Machine Company of Baltimore.

In April 1903 Mr. Sirich became an assistant engineer on the steamships of the Atlantic Transport Company, remaining with them a year, during which he secured a United States license as second assistant engineer of ocean condensing steamers of 10,000 gross tons. From 1904 to 1908 he was connected with the American Bridge Company at Ambridge, Pa., and the Westinghouse Machine Company, East Pittsburg, Pa. In the latter company he entered first the turbine-testing department, of which he became foreman in September 1905, and later was transferred to the turbine-erecting department of the New York district, as trouble foreman.

In July 1908 he became connected with the power department of the Bethlehem Steel Company and held the position of chief draftsman at the time of his death. Mr. Sirich was accidentally drowned in the Lehigh River at Bethlehem, Pa.

FREIGHT TRAIN RESISTANCE

ITS RELATION TO AVERAGE CAR WEIGHT

By Prof. Edward C. Schmidt, Urbana, Ill.

Member of the Society

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Train resistance varies not only with the train speed, but also with the average weight of the cars of which the train is composed. At a given speed the tractive effort required for each ton of weight of the train will be greater, for example, for the train which is composed of cars of twenty tons average gross weight, than for the train composed of cars which weigh, on the average, fifty tons each.

2 While this fact has been known for some years, it has found inadequate expression and but little application. In the establishment of their tonnage ratings many railroads have altogether ignored it. In the tonnage ratings of a few roads this variation of resistance with car weight is recognized to the extent of allowing a difference in rating between trains composed of loaded cars and those consisting entirely or partially of empty cars. Generally in such systems a certain amount is allowed arbitrarily, to be added to the weight of empty cars in determining, for the purpose of rating, the weight of the train in which they are found. In such rating no distinction is made between loaded cars of various weights, although such weights vary from 25 to 70 tons.

The American Society of Mechanical Engineers, 29 West 39th Street, New York. All papers are subject to revision.

3 A still smaller group of railroads have fully recognized the significance of the facts above stated in establishing their tonnage ratings, which in such cases are usually termed "adjusted" or "equated" ratings. Under these adjusted ratings the actual weight of the train allotted to a particular locomotive varies according to the number of cars in the train. The rating for the same locomotive, with trains of 40, 60, and 80 cars, for example, will be different in each of the three cases. This, of course, is in effect a variation of the rating with respect to the average car weights.

4 Most of these adjusted ratings have been empirically determined. In the few cases where they rest upon experiments made to determine the variations in train resistance with respect to car weight the data and results of such experiments have not been fully published. Existing train resistance formulæ likewise fail in most cases to take into account these variations of resistance with car weight; and probably much of the divergence among them is properly to be ascribed to this fact.

5 *Purpose of the Tests.* In view of the facts just stated it has seemed desirable to make the tests whose results are here recorded. They were planned to determine the resistance of freight trains under the usual conditions of operation; and were designed to disclose at the same time, if possible, the relation existing, at any given speed, between train resistance and average car weight. Since the chief use of such information is in the production of locomotive ratings, the conditions of the tests have been made like those which prevail in normal freight train operation. The speed range, for example, is from 5 to 35 miles per hour; and the trains experimented upon were trains in regular service, and usual in their make-up. The track upon which the tests were made is believed to be representative of good main line construction.

6 The tests have been made as part of the research work of the Engineering Experiment Station of the University of Illinois, conducted by the railway engineering department. They were begun in April, 1908, and were completed in May, 1909. All tests were made by means of Test Car 17, a dynamometer car, owned jointly by the University of Illinois and the Illinois Central Railroad, and were carried out on the Chicago division of this road.

7 In the preparation of the report the aim has been to present in it as brief a statement of the results and conditions as is compatible with a clear understanding of the tests. The results of the tests will also be published as a bulletin of the Engineering Experiment Station

of the University of Illinois. This bulletin will contain, in addition to the facts here presented, more detailed information concerning the track, the dynamometer car, and the methods of calculation; as well as the tonnage record for each train and the results and resistance curve for each test. A summary of the test conditions and the conclusions is inserted immediately beyond.

8 Throughout the report the terms "resistance" and "train resistance" mean the number of pounds of tractive effort required for each ton of the train, in order to keep it in motion on straight and level track, at uniform speed, and in still air. The report deals exclusively with the resistance of the train behind the locomotive tender. Locomotive and tender resistance are not discussed.

9 *Acknowledgments.* The tests have been made possible through the interest and coöperation of Mr. William Renshaw, Mr. J. G. Neuffer, and Mr. R. W. Bell, who were successively superintendents of machinery of the Illinois Central Railroad during the period of planning and conducting the work. Many other officials of the Chicago division of the road have rendered generous assistance in the investigation, which has entailed for them not a little inconvenience and labor. Such interest and assistance are thoroughly appreciated by those of the university staff who have been concerned with the work.

10 Throughout this work, the operation of the dynamometer car and the making of the calculations have been under the direct supervision of Mr. F. W. Marquis, Associate in the railway engineering department, Engineering Experiment Station. Much of whatever accuracy and reliability have been attained in the investigation is due to his intelligent and painstaking care in making the tests and in systematizing the work of calculation. He has also rendered great assistance in supervising the preparation of the tables and illustrations, and in the final checking of the manuscript.

SUMMARY AND CONCLUSIONS

11 *Summary.* The report deals with the results obtained from tests of thirty-two ordinary freight trains, whose chief characteristics were as follows:

	Minimum	Maximum
Total weight, tons	747	2908
Average weight per car, tons	16.12	69.92
Number of cars in the train	26	89

The trains whose average weights were less than 20 tons or more

than 60 tons were composed of cars of nearly uniform weight; while those whose average car weights were between 20 and 60 tons were either homogeneous or mixed as regards the weight of the individual cars.

12 The tests were made during generally fair weather. The minimum air temperature during any test was 34 deg., the maximum 82 deg. The approximate average wind velocity prevailing throughout one test was 25 miles per hour; during all the others it was less than 20 miles per hour.

13 The tests were made upon well constructed and well maintained main-line track, 94 per cent of which is laid with 85-lb. rail, the remainder being laid with 75-lb. rail. Except through station grounds, where screenings or cinders are used for ballast, the track is full ballasted with broken stone.

14 *Conclusions.* The results of the tests are presented in Figs. 10 and 11, in Table 3, and in the equations accompanying Par. 75. The curves, the table, and the equations, are each different expressions of the same facts. It is believed that by their use one may safely predict the probable total resistance of *entire* freight trains at various speeds, when running upon straight and level track of good construction, during weather when the temperature is above 30 deg. fahr., and the wind velocity not more than 20 miles per hour; provided the *average* weight of the cars composing the train be known.

15 The results are applicable to trains of all varieties of make-up to be met with in service. They may be applied, without incurring material error, to trains which are homogeneous and to those which are mixed, as regards individual car weight.

16 The results are primarily applicable to trains which have been for some time in motion. When trains are first started from yards, or after stops on the road of more than about twenty minutes duration, their resistance is likely to be appreciably greater than is indicated by the results here presented. In rating locomotives no consideration need be given this matter, except in determining "dead" ratings for low speeds, and then only when the ruling grade is located within six or seven miles of the starting point or of a regular road stop.

17 It is to be expected that some trains in service will have a resistance about 9 per cent in excess of that indicated by Figs. 10 and 11, due to variations in make-up or external conditions within the limits to which the tests apply. If operating conditions make it essential to reduce to a minimum the risk of failure to haul the allotted tonnage, then this 9 per cent allowance should be made. This con-

sideration, like the one preceding, is important only in rating locomotives for speeds under 15 miles per hour. At higher speeds, the occasional excess in the resistance of individual trains will result in nothing more serious than a slight increase in running time. It should be emphasized that this allowance, if made, is to be added to the resistance on level track—not to the gross resistance on grades.

THE METHODS AND MEANS EMPLOYED IN CONDUCTING THE TESTS

18 The tests were carried on by means of the dynamometer car here referred to as Test Car 17, which, when not in use, is held at Champaign, a district terminus. The car was operated from time to time in the regular trains leaving this point, and the trains selected were partly in the northbound, partly in the southbound traffic.

19 The plan was to determine, for each of the trains experimented upon, the relation of its resistance to its speed. This information was to be finally expressed as a resistance-speed curve such as is shown in Fig. 1. The trains were so selected that their average car weights varied through as great a range as possible. As will later appear, this range proved to be from the weight of an empty gondola to that of a fully loaded car of 100,000-lb. capacity. It was the expectation that when the resistance-speed curves of the individual tests were brought together, their analysis would reveal the relations existing between train resistance and car weight.

20 During each test the following information was obtained:

a The drawbar pull of the locomotive upon the train.
b The train speed.
c A continuous record of time elapsed from the beginning of the test.
d The pressure existing in the brake cylinder of the test car.
e The direction of the wind relative to the direction of motion of the car.
f The velocity of the wind relative to the car.
g A record of the location of the test car upon the road.
h Air temperatures and other weather conditions.
i Data concerning the train, such as its weight, etc.

The information cited under Items *a* to *g* was obtained in the form of continuous graphical records upon the chart which is produced by the apparatus of the dynamometer car. By means of this chart any of the quantities mentioned may be determined at any point upon the road.

21 The curves of drawbar pull and speed provide the informa-

tion essential to the investigation. Supplemented by an accurate profile and a record of train weight, they enable net train resistance to be calculated at any position of the train upon the road. The time record provides a means of calibrating and checking the speed curve. The pressure in the brake cylinder was recorded merely to make it possible to distinguish those periods during the test when the brakes were applied to the train; it being obviously necessary to ignore such portions of the record when making the calculations.

22 The relative wind velocity and relative wind direction were obtained by means of an anemometer and a wind vane mounted on the roof of the test car. When compounded with the known speed and direction of motion of the car, these data permit the determination of the actual wind direction and wind velocity with respect to the track, which were recorded in each test for each point at which train resistance was determined. It is probable that these wind data are, under some circumstances, subject to a considerable error. Considering the length of the run made with each train and the length of time it was on the road, it is believed the data are, nevertheless, more reliable than those which might have been recorded by stationary instruments located at one or two points along the track.

23 Item g—the location of the car upon the road—was defined by marking upon the test-car record the position of mile posts and stations at the moment they passed the car. By means of this record it is possible to correlate any position of the train with the road profile.

24 Data concerning the train were obtained by one or two observers who had no other duties. With the one exception noted beyond, all trains were weighed to determine their tonnage. In addition to its weight, there was recorded for each train, its length[1] and for each car its number, kind, stenciled "light weight," gross weight, capacity, and the initials of the owning road.

25 All test-car instruments were calibrated before the tests, and their calibrations were frequently checked during the progress of the investigation. All observers were men experienced in the operation of the test car, and many of them had participated also in the work of calculation, and were consequently aware of the points at which alertness and care were especially needed. No effort has been spared, in conducting the tests, to ensure accuracy in the data. These facts

[1] Train length was determined by counting, during the test, the number of rail lengths corresponding to the length of the train and multiplying this number by 30 ft., which is the rail length for this track.

are here mentioned as having some significance to one who may undertake to estimate the reliability of the results. The Appendix contains an illustration of one of the test-car charts and a description of the car itself.

26 This report includes the data and results from tests of thirty-two different trains. For the purposes of this research, tests were made of twelve other freight trains; but their results were finally excluded from the report. Three of these additional tests were rejected because of uncertainty about the train weights, one because of a breakdown in the test-car recording apparatus during the progress of the test, and eight were disregarded because the temperatures prevailing were below the range for which it was intended the results should apply: the low temperature in some cases being coupled with high wind.

TEST CONDITIONS AND TRAIN DATA

27 *The Trains Tested.* The test trains were all of such make-up as naturally resulted from the traffic conditions in the Champaign yards. For most of the tests the test car was simply coupled into the trains selected by the trainmaster, solely with reference to his convenience in operating and in returning the test car. As the investigation progressed, it became apparent that the accumulated data left certain gaps in the range of average car weights. There were at this stage, for example, few trains experimented upon with average car weights near 25 to 30 tons, and none with an average car weight of 70 tons. The last six or eight trains were therefore made up especially to supplement the data at these points. It should be understood, however, that nothing in this process resulted in a train-make-up which was in any respect unusual. All the trains tested are, therefore, such as one might expect to find upon any road where the traffic conditions are normal. They include trains made up almost entirely of empty gondolas,[1] others with considerable variation in both load per car and kind of car, and still others composed almost entirely of loaded box cars or of loaded gondolas.

28 Test S-1018 demands special mention in this connection. The train for this test included Illinois Central Railroad locomotives No. 423 and No. 732, weighing respectively 145,200 and 223,600 lb.

[1] Cars are designated as box, stock, gondola, flat and tank cars. The term box car is made to include refrigerator cars, the test car and the caboose. The term gondola includes all unroofed cars with sides, such as coal cars, hopper cars, etc.

Their combined weight constituted 13.6 per cent of the total train weight. These locomotives, with their tenders, were being hauled "dead" and had the main rods disconnected, as is usual in such cases. The first is of the 2–6–0 type, the second of the 2–8–0 type, and they and their tenders had therefore together seventeen axles in operation. For the purpose of determining the average car weight for this train, these two locomotives were assumed to be equivalent, in their resistance, to a number of cars having a like number of axles, i.e., $4\frac{1}{4}$ cars. The results of the calculations warrant the belief that this view of the situation has resulted in no material error. A study of Table 1 will make clear the diversity in the composition of the trains.

29 All trains, except Nos. S-1016, S-1018, S-1030A and S-1030B, were weighed upon one of the two track scales at Champaign. This weighing was done in the usual manner, by pulling the train over the scales and weighing the cars successively without uncoupling. These track scales were in good condition and were each inspected four times during the test period, these inspections disclosing a maximum error in one scale of $-\frac{1}{5}$ per cent, in the other of $-\frac{1}{2}$ per cent. The train in Test S-1016, composed entirely of empty cars, by an error in arrangement left the yards without being weighed. The weights stencilled on the cars were accepted as correct in this case. The train in Test S-1018 was weighed upon track scales in the Chicago yards; and the trains of Tests S-1030A and S-1030B were weighed in the yards at Centralia. In Test S-1021, after leaving the yards, two cars were added to the train, for which the weights were determined from the stencilled weights and the way-bills. In Tests S-1030B and S-1048 the weights of one and two cars respectively were similarly determined, and in Test S-1061 the stencilled weight was used for one empty car. Obviously no important errors in the total tonnage have resulted from possible inaccuracies in the weights of these cars.

30 All cars of all trains were of course provided with the usual four-wheeled truck. Presumably the majority of the cars had journals conforming to the specifications of the Master Car Builders' Association, which for some years have required that freight car journals be either $3\frac{3}{4}$ in. by 7 in., $4\frac{1}{4}$ in. by 8 in., 5 in. by 9 in., or $5\frac{1}{2}$ in. by 10 in. in size, depending upon the car capacity. It is safe to assume that all trucks were provided with wheels of 33 in. standard diameter.

31 Throughout each test, observations were repeatedly made to discover such irregularities as hot journal boxes, brakes which were not free from the wheels, and trucks which did not freely follow the track. Such things occurred to the usual extent; a hot-box or two or

an unreleased brake being occasionally found on some of the trains, while others were entirely free from such defects. The record of such matters was given consideration in making the calculations; but, as was anticipated, the results showed no discrepancies which could be explained by such causes.

32 The range over which the train data for all of the tests varied is as follows:

	Minimum	Maximum
Total train weight, tons	747	2908
Average weight of cars composing the train, tons	16.12	69.92
Number of cars in the train	26	89
Train length, ft	1120	3480

33 *The Track.* The track upon which the experiments were carried on extends from Gilman to Mattoon, Ill., a distance of 91 miles, and lies upon the Chicago division of the main line of the Illinois Central Railroad. Until about 10 years ago this was a single-track road and one of the oldest in the State. At that time a second track was constructed, and the roadbed for both tracks is now well settled and in good condition. The maximum grade against northbound traffic is 29 ft. per mile, and against southbound traffic, 31.9 ft. per mile. In all the 91 miles there are only 7850 ft. of curved track.

34 Through station grounds the tracks are ballasted with screenings or cinders; all other portions of both tracks (about 83 of the 91 miles) are full ballasted with broken limestone. The cross-ties are of oak, laid 20 in. center to center. About 10½ miles of the west track is laid with 75-lb. A.S.C.E. rail, put down in 1894 and 1895; while the remainder of the west track and all of the east track is laid with 85-lb. A.S.C.E. rail, the oldest of which was put down in 1900. During 8 months of the year there is employed in maintaining this portion of the road a force of men which averages one man per mile of track; during the other 4 months this force is reduced to one man for each 2 miles. As regards both construction and maintenance this track is such as one may expect to find upon the main lines of first-class railroads.

35 These 91 miles of track were especially surveyed, immediately preceding the tests, by the railway engineering department of the university for the purposes of this and similar investigations. The levels were run on the east track, and readings were taken to 0.1 ft. at stations 300 ft. apart; and turning points were taken at every fourth

TABLE 1 SUMMARY OF TEST CONDITIONS AND TRAIN DATA

		Weather Conditions						Train Data†										
			Air Temp. Deg. F.			Range of Direction of Wind with Respect to Track*			Weights			Train Make-up						
												Conditions of Loading				Kind of Cars		
Test Lab. Serial No.	Test Date		Beginning of Test	End of Test	Ave. Approx Wind Velocity Miles per Hour	From	To	Train Length Feet	Gross Train Weight Tons	Average Gross Weight per Car Tons	Cars in Train Total No.	Empty Cars No.	Loaded Cars No.	Loaded Cars %	Box Cars %	Gondola Cars %	Flat Cars %	Tank Cars %
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19
	1908																	
S-1013	4/27	Wet	42	44	19	+35°L	+90°L	2784	2549	38.04	67	10	57	85	82	13	0	5
S-1015	4/29	Fair	40	48	10	+25°L	−80°L	2520	2489	36.08	69	8	61	88	68	6	16	10
S-1016	4/30	Fair	44	48	10	+15°R	+60°R	3030	1161	16.12	72	72	0	0	3	97	0	0
S-1017	5/1	Wet	48	54	16	+45°L	−80°L	2670	2532	38.44	66	13	53	80	95	5	0	0
S-1018‡	5/2	Fair	40	45	11	+10°R	+85°R	2130	1353	25.40‡	49‡	34	15	31	61	6	16	17
S-1019	5/9	Fair	44	62	25	+20°R	+45°R	3480	1572	17.72	89	75	14	16	34	58	7	1
S-1021	5/13	Wet	66	70	17	+60°R	+80°R	2400	2908	46.16	63	10	53	84	32	60	5	3
S-1023	5/23	Fair	62	74	17	+75°R	−80°R	2320	2243	38.72	58	17	41	71	45	52	0	3
S-1027	7/2	Wet	64	80	14	−50°R	+70°R	1710	2185	47.44	46	3	43	94	22	76	2	0
S-1030A	7/8	Fair	60	68	6	+20°R	+65°R	1380	2036	59.88	34	2	32	94	6	94	0	0
S-1030B	7/8	Fair	68	72	7	+20°R	+65°R	1650	2342	57.12	41	3	38	93	20	80	0	0
S-1031	7/22	Fair	70	82	5	+ 0°	+40°L	1425	747	20.72	36	30	6	17	94	3	3	0
S-1032	9/26	Fair	66	82	12	+ 5°L	+15°R	1710	2275	51.70	44	2	42	95	5	95	0	0
S-1034	10/3	Fair	42	60	4	− 0°	+85°L	3015	1259	16.56	76	76	0	0	1	99	0	0
S-1036	10/10	Fair	40	62	6	+15°R	−15°R	2010	1961	37.72	52	8	44	85	73	25	2	0

S-1038	10 15	Fair	58	72	16	+ 5° L.	+25° L.	1650	2144	52 28	41	3	38	93	22	78	0	0
S-1040	10 24	Wet	57	53	11	+15° R	+40° R	1830	2152	45 76	47	2	45	96	49	49	0	2
S-1043	11 7	Fair	38	53	8	+ 5° R	+65° R	2580	1118	16 92	66	65	1	2	24	74	0	2
S-1048	11 28	Fair	36	39	6	+ 5° R	+30° L	A2175	2443	45 24	54	8	46	85	37	63	0	0
								B2100	2355	45 24	52	8	44	85	38	62	0	0
	1909																	
S-1050	1 23	Fair	53	66	8	+ 0°	−25° R	1620	1618	40 44	40	16	24	60	75	25	0	0
S-1052	1 28	Wet	36	40	11	−45° L	+70° L	2430	1514	24 80	61	44	17	28	61	38	1	0
S-1057	2 6	Fair	34	40	16	+20° R	−35° L	1830	2107	41 32	51	8	43	84	49	43	6	2
S-1061	3 13	Fair	41	38	7	+45° L	−85° L	1785	2252	51 20	44	3	41	93	5	84	11	0
S-1063	3 19	Wet	36	40	12	+20° R	+40° R	3060	1484	20 04	74	70	4	5	7	93	0	0
S-1070	4 17	Fair	58	71	4	+ 0°	−65° L	2400	1622	24 60	66	49	17	26	58§	42	0	0
S-1072	5 1	Fair	35	37	17	+70° L	+90° L	1200	1859	66 40	28	1	27	96	4	96	0	0
S-1073	5 4	Fair	53	63	10	+25° L	+70° R	1200	1880	67 16	28	1	27	96	4	96	0	0
S-1074	5 7	Fair	45	60	10	+65° L	−80° L	3180	1340	16 56	81	81	0	0	2	98	0	0
S-1076	5 11	Fair	51	67	16	+40° R	+75° R	1120	1818	69 92	26	1	25	96	4	96	0	0
S-1077	5 14	Fair	64	70	13	−25° R	−75° R	2145	1505	28 40	53	35	18	34	74	26	0	0
S-1079	5 18	Fair	65	68	18	+65° R	−85° R	2070	1685	33 04	51	14	37	73	90	10	0	0
S-1080	5 21	Fair	50	70	11	+ 0°	+45° L	2550	1347	21 40	63	57	6	10	16	84	0	0

* Direction is designated by the angle made with the track. A wind any component of whose velocity helps the train forward is marked +; winds with components of opposing velocity are marked −. Winds from the right side of the track are designated as R; from the left side as L. Thus +40° R. means a wind blowing from the rear and from the right hand side, whose direction makes an angle of 40 deg. with the track.

† All data apply to the train only—engine and tender are excluded. In Columns 9 to 19, for test S—1048, A indicates, from Champaign to Rantoul. B from Rantoul to Gilman.

‡ Train in Test S-1018 had two "dead" locomotives and tenders in addition to cars noted.

§ This number included 15 stock cars—classed as box.

station where levels were read to 0.01 ft. The results of the survey are expressed in a profile drawn to a scale of $\frac{1}{4}$ in. to 100 ft., which was used in making the test calculations.

36 *The Weather Conditions.* In Table 1 the weather prevailing during each test is designated as either fair or wet, wet weather meaning either continuous or intermittent rain. During seven of the thirty-two tests the weather was wet. The lowest air temperature recorded at any time during any test is 34 deg. fahr. and the highest recorded temperature is 82 deg.

37 The column headed "average wind velocity," in Table 1, presents the averages of the calculated wind velocities derived for each point or section of the test in question for which the train resistance was determined. There was a considerable variation at different points during the same test. The approximate maximum average velocity prevailing during any test was 25 miles per hour, the minimum 4 miles per hour. The actual wind direction (with respect to the track) varied during the tests, as would be expected, through the entire 360 degrees.

38 It was intended so to select the tests that the weather conditions, the temperatures, and the wind velocities, would be such as usually prevail in most parts of the country from the middle of spring until the middle of autumn when the basic or "summer" tonnage ratings are in force—such conditions, in short, as would give rise to no appreciable difficulties in train operation.

METHODS EMPLOYED IN CALCULATING THE RESULTS

39 The immediate purpose in making the calculations was to produce for each test a curve showing the relation between resistance and speed, for as great a variety of speeds as the data would permit. This involves calculating the train resistance at various positions of the train upon the track, and the first step is the inspection of the test-car record in order to select suitable points or sections at which the resistance may be calculated. The considerations of first importance are, that the points should finally represent as great a speed range as possible, and that the speeds should be approximately evenly distributed within this range. Only points and sections where the entire train was running and continued to run upon straight track, were selected; resistance due to track curvature is therefore entirely eliminated. The data essential to the process of calculation are the drawbar pull of the engine, the train speed and its acceleration, the ton-

nage, and the profile. The pull and the speed, as previously stated, were determined from continuous curves drawn on the test-car chart. Two processes were used, designated here as Method 1 and Method 2. By Method 1 the momentary values of pull, speed, acceleration, and grade, were determined for a particular position of the train upon the road; by Method 2 the average values of these quantities were determined for the period during which the test car was passing over a definite section of the track.

40 *Method 1, Resistance at a Point on the Road.* The point having been chosen, the pull and the speed were found by direct readings from the chart. This pull divided by the tonnage gives the gross train resistance at this speed. This gross resistance was next corrected for both acceleration and grade resistance. The acceleration was determined, by graphical methods from the speed curve, and the grade was found by correlating the train's position with the profile. The points were all so selected that at the moment under consideration, the entire train was on a nearly uniform grade. Method 1 results in momentary values of train resistance at the points considered.

41 *Method 2, Average Resistance over a Section.* By this method the average value of train resistance was determined for the period during which the test car at the head of the train was passing a selected section of the track. This track section, corresponding to a certain length or section on the test-car record, was so selected that the speed of the car when entering was nearly equal to its speed when leaving it; and further so that no considerable variations in speed occurred during transit over the section. The sections chosen have varied in length from about $\frac{1}{4}$ mile to 1 mile. The variations in speed in passing the section have generally amounted to less than 2.0 miles per hour, and the maximum variation over any selected section is 11.7 miles per hour. In only 58 cases out of a total of 560 does this speed variation exceed 5.0 miles per hour.

42 These portions of the chart being chosen, the average pull was next found by determining the average ordinate of the curve of drawbar pull, and the average speed was found by means of the section length and the time record. Gross resistance in pounds per ton was next derived by dividing this value of pull by the tonnage, and this gross resistance was then corrected for the resistances due to acceleration and grade as in Method 1.

43 In this case the average acceleration was found by consideration of the speeds at entrance to and exit from the section. In order

to correct for grade, the elevation of the center of gravity[1] of the train was determined for that position of the train at which the test car entered the section, and again for the position at which the car left the section. The difference between these elevations establishes the effective average grade, which either helps or opposes the locomotive while the train passes the section. These elevations of the center of gravity of the train may not be determined with sufficient accuracy unless the train at the moment, is on a practically uniform grade. The section limits were therefore so chosen.

44 Method 2 results in a value of *average* train resistance for the *average* speed at which the train passes the section under consideration. It would be rigidly correct if train resistance varied uniformly with speed, in other words, if the curve showing the relation of resistance to speed were a straight line. This, of course, is not the case, and the process therefore gives results which are slightly in error. However, as stated above, the section was so chosen that the difference between the speeds at entrance to and exit from the section was small; and for the speed range represented by this difference, the curve of train resistance deviates but little from a straight line. Such error as does result from the process is, therefore, very small and of no moment whatever when compared with the variations due to natural causes, that occur in the resistance itself.

45 *Comparison of the Two Methods.* The two methods are fundamentally alike. Although the first is the less laborious, it requires the determination of acceleration at a point on the speed curve; which it is sometimes difficult to make accurately. For this reason the second method is generally preferable. Method 2 also deals with average values and therefore tends to eliminate from the results the incidental momentary variations in resistance. Consequently the second method has been employed whenever possible, and the first method generally resorted to only in cases where the limitations imposed in the selection of sections for Method 2 would have resulted in too few values from which to plot the resistance curves. Of all the individual resistance values incorporated in the report only 32 per cent were determined by Method 1. The care exercised in the calculations, and a study of the plotted values obtained by both processes, seem to

[1] The location in the train of its center of gravity was determined thus: Assume a train which weighs 1800 tons, is 2400 ft. long, and is composed of 60 cars. By inspection of the tonnage record we find that one-half of this weight (900 tons) lies in the first 25 cars. Hence the center of gravity is located $\frac{25}{60} \times 2400 = 1000$ ft. from the front end.

warrant the conclusion that their results are equally reliable. In Fig. 1 the circles represent values derived by Method 1, and the circular black spots represent values obtained by Method 2.

46 *General Considerations.* Even in freight train operation the tractive effort required to produce acceleration in speed is frequently greater than that required to overcome all other resistances combined. To produce, for example, an acceleration of 0.1 miles per hour per sec., requires a tractive effort of about 9 lb. per ton, in addition to that required by net train resistance and grade resistance. Since the acceleration resistance may constitute so large a proportion of the gross resistance, it is important that its determination be made with great care. This fact has been impressed upon all concerned with these tests. In calculating the acceleration resistance, both the force required to produce acceleration in the rotation of the wheels and axles, and the force required to produce the acceleration in the motion of translation of the train as a whole, were determined.

47 The test-car records make it possible to distinguish those portions of each test where the brakes were applied. Such places, few in number, were of course avoided in selecting points and sections for determining resistance. The records also show where hot-boxes and unreleased brakes were discovered in the train, and such defects were given consideration in making the calculations. They occurred infrequently and their effect could not be distinguished in the results. While therefore such portions of the record were avoided if convenient, sections and points on the charts otherwise suitable for calculation, were not rejected on these accounts.

48 *Effect of Stops in Limiting the Selection of Points and Sections.* Early in the progress of this work, when low air temperatures were first encountered, it became apparent that when the train was first started from rest, its resistance, calculated for a number of points at which the speed was the same, occasionally was unusually high. This was true not only for those portions of the run made immediately after leaving the yards; but also for those portions immediately following stops on the road. In a certain test, for example, the values of net resistance, calculated at various points at all of which the speed was 20 miles per hour, varied between 6.8 lb. and 5 lb. per ton—a difference of 27 per cent—for points selected within the first 9 miles of the run; whereas values of resistance at the same speed, determined later in the test, differed by only 10 per cent. The air temperature during this test (not included in the report) varied between 22 deg. and 26 deg.

49 For a number of tests such resistance values were plotted with respect to the distances from the yards of the points to which they apply. This process disclosed a surprisingly regular decrease in the resistance, until a distance of approximately ten miles was reached, after which the resistance had settled down to a fairly uniform value. Similar variations were found to occur to some extent during tests when the air temperature was as high as fifty or sixty degrees. This study[1] led to the conclusion that the difference in resistance was due to variations in the conditions of lubrication of the car journals, and that such variations were chiefly caused by changes in journal temperature. All this is, of course, in accord with the common belief of those experienced in train operation. The reason for discussing it in this place is that the facts stated have influenced the procedure in making calculations for this series of tests.

50 Since the variations in resistance are so great during the early part of the run no point or section within about the first ten miles has been selected for calculation in any test. If other points or sections, located farther from the start, were near stops, such points were rejected unless further investigation proved that at these places the train resistance had become nearly uniform in value. Fortunately operating conditions were such as to entail few stops, and the selection of points and sections for the calculations has not been unduly limited on these accounts.[2]

51 The effect of these limitations is to make the results of this investigation primarily applicable to trains which have been for some time in motion. Since, however, stops are not usually made upon ruling grades; and since if stops are made at other places on the road, the locomotive has available tractive power in excess of the requirements, the results of these tests are generally applicable in the solution of tonnage rating problems, except where the ruling grade occurs near a yard or other point where the trains are made up. In such cases the tonnage determined from the resistance curves here presented may prove to be somewhat too great.

52 *The Derivation of the Resistance Curves.* The calculations

[1] Further investigation of this matter is in progress, and the results are likely soon to be published.

[2] During the thirty-two tests included in the investigation, only sixty-eight stops, all told, were made after leaving the yards. Of these, one was of 55 minutes duration, nine lasted between 20 and 40 minutes, twenty-two between 10 and 20 minutes, and thirty-six less than 10 minutes.

result, for each test, in a series of values of net train resistance at a variety of speeds. These values of resistance were plotted with respect to speed, and gave such a diagram as is shown in Fig. 1. The curve, such as is there shown, was next drawn to express, for the test in question, the relation existing between resistance and speed. In order to draw this curve, the plotted points were assumed to be arranged in a number of groups, and for each group the averages of the values of speed and of resistance were determined. By these averages a new point or "center of gravity" of the group was then plotted. The curve was drawn by confining attention to the few points thus determined. The groups of points were arbitrarily selected so that the resulting "centers of gravity" would be nearly equidistantly distributed throughout the speed range. All curves presented in the report, except those exhibited in Fig. 11, were drawn by this process.

53 All reasonable precautions have been taken to attain accuracy in the calculations. In determining each value of resistance, each step was duplicated at a different time, and generally by a different person. The transcription of all tables, the plotting of points and the drawing of curves, have been similary checked.

RESULTS OF THE TESTS

54 *Results of the Individual Tests.* The immediate result of each test is a curve which shows for the train under consideration, the relation existing between train resistance and speed. Fig. 1 is such a curve derived from test S-1021. It is fairly representative of the entire group of curves, and such discussion of it as follows is general in its application.

55 The plotted points show unmistakably an increase in resistance as the speed increases, and the curve drawn represents the mean relation between resistance and speed. In Fig. 1 the maximum variation from this mean of any calculated value of resistance is about 20 per cent; the next largest variation is 16 per cent, and other calculated values of resistance generally differ from the values determined from the curve by less than 10 per cent. In a majority of the tests the maximum variation is less than in Fig. 1, and the general agreement between the calculated values of resistance and the ordinates of the curve is better than in the test chosen for illustration.

56 It has been thought desirable to express more specifically this variation between the calculated values of resistance and the mean values as derived from the curves drawn. To this end, for all tests,

all calculated values of resistance for speeds between 8 and 12 miles per hour were compared with the ordinates of the curves at the corresponding speeds, and the percentage difference was determined in each case. These percentages were then arranged in two groups and averaged. The one group included the results from all points lying above the curve, the other from those lying below it. The whole process was next repeated for speeds between 28 and 32 miles per hour, with results as follows:

AVERAGE DEVIATION (FOR ALL TESTS) OF CALCULATED RESISTANCE FROM THE MEAN VALUES DERIVED FROM THE CURVES, EXPRESSED IN PERCENTAGE OF THE MEAN VALUES.

Speed	Above the Mean	Below the Mean
8 to 12 m.p.h.	6.4 per cent	7.6 per cent
28 to 32 m.p.h.	5.6 per cent	6.6 per cent

Such variation seems not extraordinary for this class of experimental work.

57 These differences may be due in part to accumulated errors in the instruments or in the calculations. In all cases, however, where the calculated value of resistance varied by an unusual amount from the mean, all calculations leading thereto were repeated and errors thus discovered are eliminated from the report. The explanation for such differences need not be sought further than in the variations which actually occur from time to time, in the resistance itself. Variations in such components of train resistance as flange friction and wind resistance are probably sufficiently great to account for the differences discussed above. The data do not permit the influences of such components of resistance to be differentiated.

58 The curve drawn for each test has been accepted as representing the average values of net train resistance, with a degree of accuracy sufficient for the purpose of rating locomotives. Such temporary excess of resistance as may be expected to occur will generally be absorbed in that reserve in the tractive effort of the locomotive which must be allowed in any system of tonnage rating.

59 *Results of all the Tests.* The resistance curves for the individual tests have all been brought together on one sheet, a reproduction of which is shown as Fig. 2.* Fig. 2 displays the immediate results

* The numbers shown on the curves are the last two figures of the test numbers. The curve marked 43 is derived from Test S-1043.

of the whole research. The lower curves give values of resistance varying from 3 to 5½ lb. per ton, while the upper curves show resistance values varying from 7 to 14 lb. per ton. Resistance values at the lower speeds differ by 100 per cent, and values at higher speeds differ by as much as 200 per cent.

60 If further analysis had not revealed the cause of the great variation in resistance here shown, Fig. 2* would have remained a useless exhibit. The explanation of this variation was sought in the test conditions enumerated below:

a Weather and temperature conditions.
b Wind velocity and direction.
c Kind of cars composing the train.
d Position of the loaded cars in the train.
e Defects in train equipment.
f Average weight of the cars in the train.

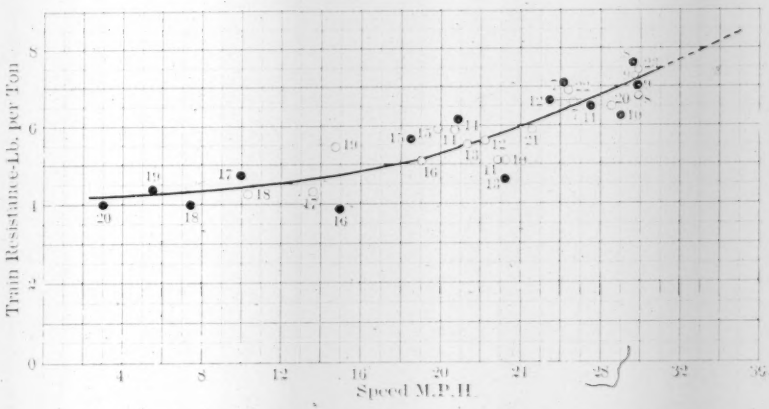

FIG. 1. RELATION OF RESISTANCE TO SPEED FOR TEST S-1021
AVERAGE WEIGHT PER CAR 46.16 TONS

61 The first five conditions either are uncontrollable or purposely were not controlled during these experiments, and attempts to explain the differences by reference to one or the other of them have been altogether unsuccessful. While difference in wind velocity, for example, might be a plausible explanation of the differences between two or

*Table 2 has been prepared from the original curves of the individual tests, only one of which is separately presented in the report (see Fig. 1). It gives no information not obtainable from Fig. 2; but presents the information in more convenient form, since the number of curves drawn in the figure makes it confusing.

three curves selected at random from Fig. 2, such explanation will not hold when applied to two or three other curves similarly chosen; and it fails altogether to explain such differences when it is applied to the whole group. The same remarks apply to any other of the first five items.

62 Item *f*, however, has furnished the clue whereby the apparent confusion in the results of the tests, as exhibited in Fig. 2, has been explained. It may be stated at once that the difference in train resistance for various tests is believed to be chiefly due to differences in the average gross car weights existing during the tests. An explanation of the process which led to this opinion follows immediately below. As was stated at the outset, this conclusion was anticipated when the work was begun, and the average car weight was therefore controlled during the experiments, and made to vary through the widest possible range.

63 *The Effects of Car Weight on Resistance.* The four upper curves of Fig. 2 are derived from trains in which the average weight per car was about 16 or 17 tons. The lowest curves are those derived from trains in which the car weight was nearly 70 tons. These facts serve as a rough indication of the part played by car weight in effecting changes in train resistance. This influence is more definitely brought out in the following discussion.

64 If from each of the curves of Fig. 2 the value of resistance is determined at one speed, say 5 miles per hour, these values of resistance may then be plotted with respect to their corresponding values of car weight; and since the speed is common, its influence is eliminated and the resulting diagram may be expected to reveal the relation existing between train resistance and average weight per car. Table 2 was prepared to facilitate this process. In it the tests are arranged in the order of the average car weights, which are given in the second column, and in the succeeding columns are set down the resistance values obtained from the *curves* of the individual tests, for each of seven different speeds. Table 2 therefore presents the values of the coördinates of seven points on each of the curves of Fig. 2, and hence, like Fig. 2, summarizes the immediate results of all tests.

65 In Table 2 the second and third columns present a series of values of average car weight and of train resistance at 5 miles per hour. Each pair of values represents the results of one of the 32 tests. Using these pairs of values as coördinates, a series of points has been plotted to form a new diagram, Fig. 3. For example, the point marked 21 in Fig. 3 is derived from the curve of Test S-1021.

The curve of resistance for this test (see Fig. 1 or Fig. 2) shows that at 5 miles per hour the mean resistance is 4.21 lb. per ton. During this test the average weight of the cars in the train was 46.16 tons. Table 2 also exhibits both of these values which, when plotted in Fig. 3, determine the point there marked 21. The other points of Fig. 3 were similarly determined. Each point represents the value of resistance at 5 miles per hour, derived from a particular test train.

66 Although there is considerable variation among the points of Fig. 3, they indicate clearly a decrease in the resistance as the car weight increases. The curve drawn in Fig. 3 represents, for the trains tested, the mean relation which existed, between resistance at 5 miles per hour and the average car weight.[1] For higher speeds this relation between resistance and car weight is shown by Figs. 4 to 9, which were derived by the same methods employed in producing Fig. 3.

67 The variation in resistance represented by the points in Figs. 3 to 9 is sufficient to warrant further discussion. Such discussion will, however, be postponed until later in the report. The conclusion reached is that these variations are largely caused by factors which are uncontrollable in ordinary train operation. If this be admitted, it is clear that the discussion of such variations may enter into the solution of tonnage rating problems, only as an argument for reserve tractive effort in the locomotive. An estimate of the desirable amount of such reserve appears beyond.

68 The curves of Figs. 3 to 9 have been accepted as representing, for these tests, the mean relation which existed between train resistance and the average gross weight of the cars composing the trains. These curves exhibit this relation at seven different speeds, 5, 10, 15, 20, 25, 30, and 35 miles per hour. For convenience in use, and to make comparison easier, these seven curves have been brought together in one diagram which is reproduced in Fig. 10.

69 Fig. 10 presents the final results of the whole research. Each of the curves there drawn shows the mean relation which existed during the tests, between car weight and resistance at a definite speed. It is believed that the curves of Fig. 10 are generally applicable to ordinary American freight trains, provided the conditions surrounding their operation are like those which prevailed during these tests.

[1] As has been previously explained, the curve is drawn by finding the "center of gravity" of several groups of points. These centers are defined in Figs. 3 to 9 by the crosses within circles. Points 34 and 74 were virtually ignored in drawing the curves of Figs. 6 and 7. The numbers at the points are the last two figures of the test numbers.

FIG. 2. CURVES SHOWING RELATION BETWEEN RESISTANCE AND SPEED FOR EACH OF THE 32 TESTS

TABLE 2. RESISTANCE AT VARIOUS SPEEDS

DERIVED FROM THE CURVES FOR THE INDIVIDUAL TESTS. THIS TABLE PROVIDES THE COORDINATES OF THE POINTS PLOTTED IN FIGS. 3 TO 9

Test No.	Average Weight per Car Tons	Train Resistance—Pounds per Ton						
		5 m.p.h.	10 m.p.h.	15 m.p.h.	20 m.p.h.	25 m.p.h.	30 m.p.h.	35 m.p.h.
S-1016	16.12	7.35	7.40	7.62	8.37	9.91	12.22	.....
S-1034	16.56	8.10	8.70	9.92	11.90	14.30	.....	.....
S-1074	16.56	6.92	8.23	10.10	13.32	14.70	.....	.....
S-1043	16.92	8.50	8.61	8.85	9.30	10.00	10.95	12.04
S-1019	17.72	7.30	7.47	7.90	8.85	10.32	.....	.....
S-1063	20.04	6.98	7.13	7.43	7.90	8.63	9.63	.....
S-1031	20.72	.....	6.24	6.30	6.40	6.73	7.60	8.94
S-1080	21.40	4.49	5.57	6.75	7.94	9.15	10.35	11.55
S-1070	24.60	5.93	6.63	7.47	8.57	9.90	.....	.....
S-1052	24.80	7.55	7.63	7.80	8.10	8.55	9.20	10.05
S-1018	25.40	5.80	5.95	6.20	6.63	7.22	8.26	10.02
S-1077	28.40	4.32	4.91	5.58	6.34	7.15	8.01	8.96
S-1079	33.04	3.66	4.30	4.92	5.60	6.22	6.89	7.55
S-1015	36.08	5.20	5.36	5.52	5.70	6.02	6.71	7.95
S-1036	37.72	4.98	5.03	5.12	5.15	5.31	5.88	7.15
S-1013	38.04	.....	5.40	5.65	5.95	6.32	6.90	7.68
S-1017	38.44	5.90	5.95	6.02	6.20	6.48	7.01	8.03
S-1023	38.72	4.16	4.80	5.56	6.40	7.30	8.25	.....
S-1050	40.44	.....	5.10	5.25	5.40	5.62	5.90	6.33
S-1057	41.22	3.40	3.88	4.35	4.83	5.31	5.80	6.30
S-1048	45.24	4.05	4.35	4.80	5.48	6.30	7.23	.....
S-1040	45.76	4.22	4.30	4.40	4.58	4.90	5.52	6.53
S-1021	46.08	4.21	4.41	4.72	5.20	6.15	7.20	8.40
S-1027	47.44	4.31	4.48	4.67	4.90	5.22	5.79	6.55
S-1061	51.20	3.50	4.00	4.51	5.01	5.51	6.01	6.53
S-1033	51.72	4.10	4.15	4.20	4.25	4.32	4.40	4.65
S-1038	52.28	3.30	3.50	3.71	3.95	4.25	4.60	5.08
S-1030B	57.12	3.73	3.80	3.82	3.90	4.10	4.50	.....
S-1030A	59.88	3.84	3.88	3.92	4.10	4.45	4.95	.....
S-1072	66.40	3.40	3.50	3.70	4.10	4.61	5.27	6.00
S-1073	67.16	2.52	2.90	3.30	3.70	4.10	4.50	4.90
S-1076	69.92	2.97	3.13	3.37	3.70	4.04	4.49	4.95

The curves of Fig. 10 enable one to determine the probable mean resistance of any such train, at speeds between 5 and 35 miles per hour, provided the average weight of the cars composing the train be known.

70. *The Results Expressed as Resistance-Speed Curves.* While Fig. 10 presents the main results of the experiments, the form in which these results are there expressed is unusual. Ordinarily train resist-

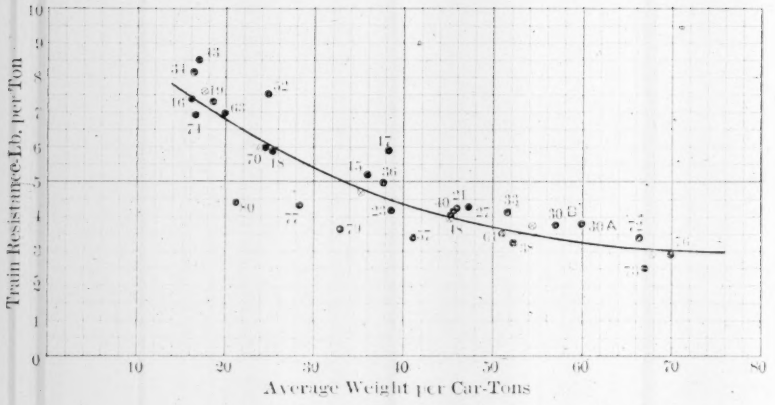

FIG. 3 RELATION BETWEEN RESISTANCE AND AVERAGE CAR WEIGHT
SPEED 5 MILES PER HOUR

ance is expressed either as a curve or equation which defines the relation between resistance and speed, instead of the relation between resistance and car weight as in Fig. 10. Obviously to express the results of these experiments in the usual form a single curve will not

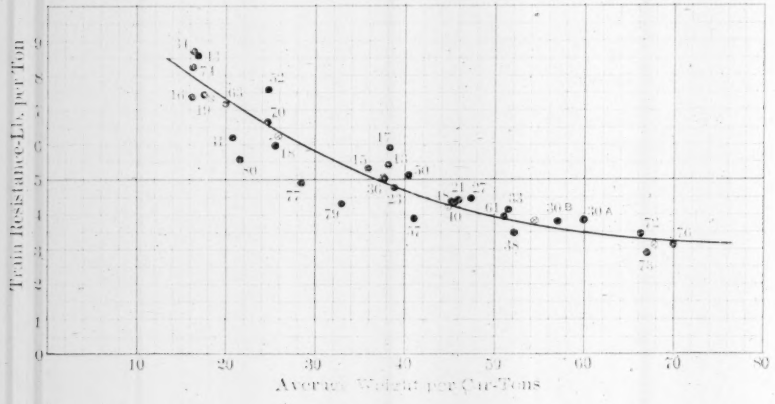

FIG. 4 RELATION BETWEEN RESISTANCE AND AVERAGE CAR WEIGHT
SPEED 10 MILES PER HOUR

suffice, since the influence of car weight cannot be thereby made evident. A number of curves will be required for this purpose, each of which will apply only to a definite average car weight. Fig. 11 presents such a group of resistance-speed curves, which have been

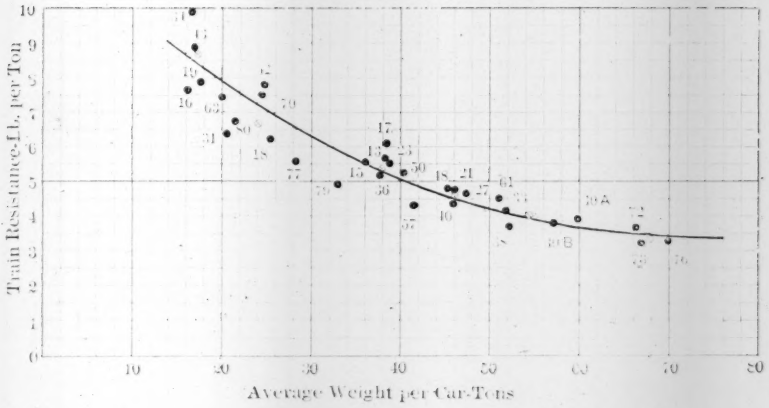

FIG. 5 RELATION BETWEEN RESISTANCE AND AVERAGE CAR WEIGHT
SPEED 15 MILES PER HOUR

derived directly from the curves of Fig. 10. Fig. 11 therefore exhibits in different form only such information as is obtainable from Fig. 10.

71 The relation between the two figures may be made clear by explaining the derivation of the upper curve in Fig. 11, the one apply-

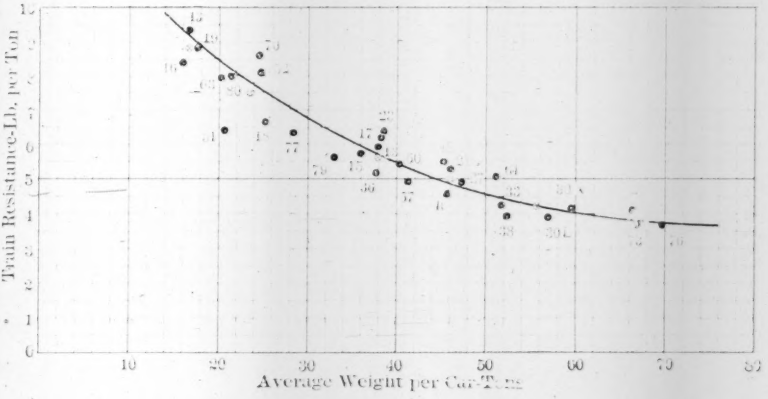

FIG. 6 RELATION BETWEEN RESISTANCE AND AVERAGE CAR WEIGHT
SPEED 20 MILES PER HOUR

ing to a car weight of 15 tons. In Fig. 10 the ordinate corresponding to an average car weight of 15 tons cuts the seven curves there drawn at seven points, at which the mean resistance values are 7.62, 8.20, 8.81, 9.56, 10.37, 11.24 and 12.25 lb. per ton, corresponding to

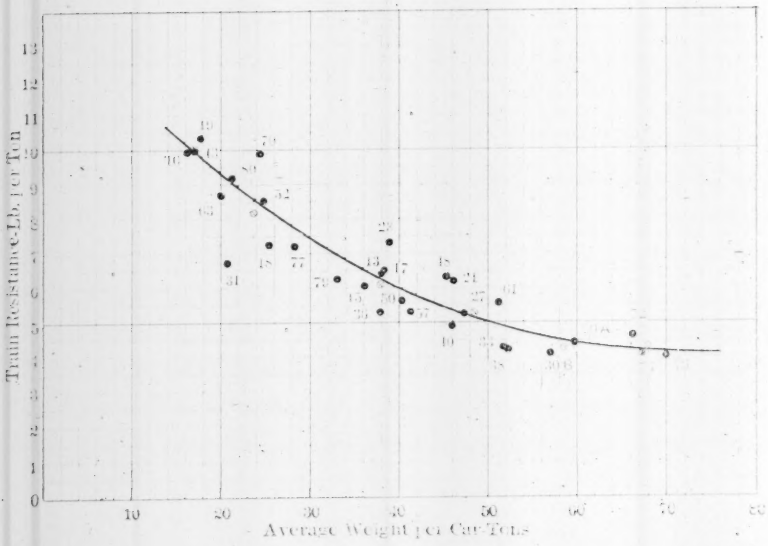

FIG. 7 RELATION BETWEEN RESISTANCE AND AVERAGE CAR WEIGHT
SPEED 25 MILES PER HOUR

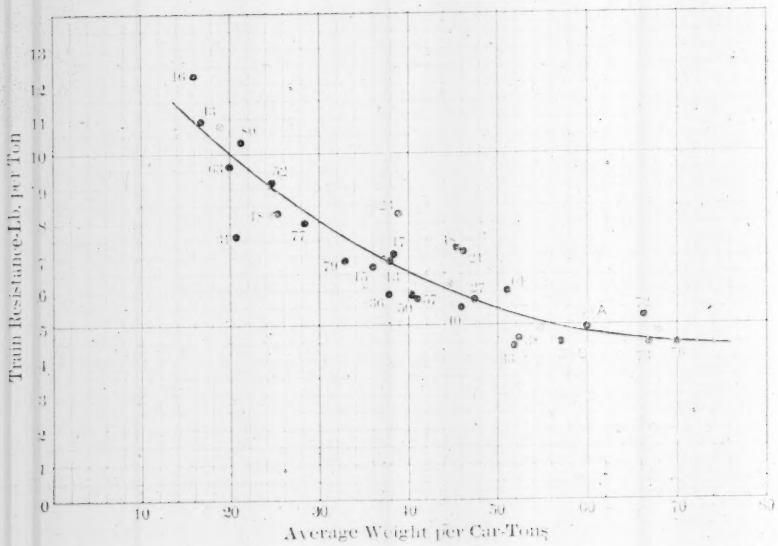

FIG. 8 RELATION BETWEEN RESISTANCE AND AVERAGE CAR WEIGHT
SPEED 30 MILES PER HOUR

speeds of 5, 10, 15, 20, 25, 30 and 35 miles per hour respectively. These values are the coördinates of seven points on a resistance-speed curve applying to a car weight of 15 tons. These seven points have been plotted in Fig. 11, and the upper curve there shown has been passed through them and extended to 40 miles per hour. The other curves of Fig. 11 were derived by a like process. In the original diagram three additional curves, corresponding to 55, 65 and 70 tons per car, were drawn. These three curves have been omitted from the figure to avoid confusion. Fig. 11 reproduces quite exactly the facts presented in Fig. 10;* and presents the final results of the experiments.

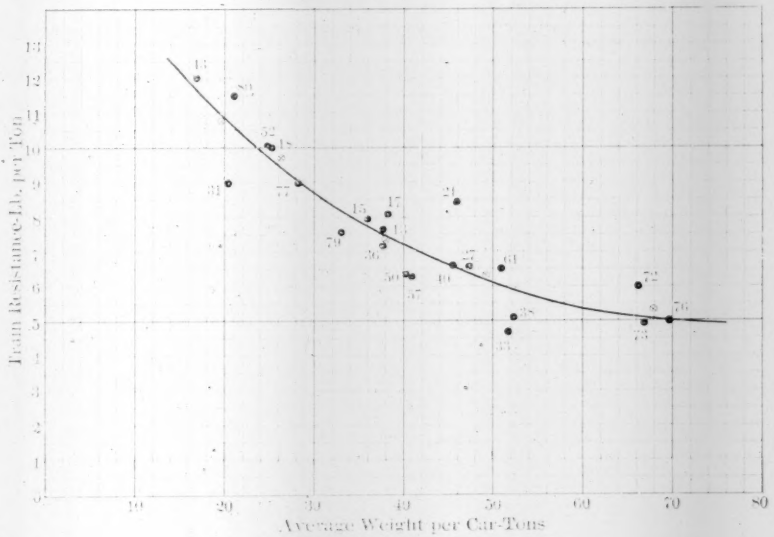

FIG. 9.—RELATION BETWEEN RESISTANCE AND AVERAGE CAR WEIGHT
SPEED 35 MILES PER HOUR

72 *The Results Expressed in Tabular Form.* From each of the curves of Fig. 11 the values of resistance at various speeds have been determined and set down in Table 3. Table 3 also includes the coördinates of the resistance curves corresponding to 55, 65 and 70 tons per car, which are omitted from Fig. 11.

*The points derived from Fig. 10 have been omitted from the tracing from which Fig. 11 was reproduced. All such points lie very close to the curves drawn in Fig. 11, the maximum deviation amounting to but $\frac{3}{4}$ of one per cent of the corresponding curve ordinate. In the Appendix there are presented tables of coördinates, by means of which Figs. 10 and 11 may be exactly reproduced.

73 *The Results Expressed as Equations.* The relation between resistance and speed shown by each of the curves of Fig. 11 may be also expressed in the form of an equation. Formulæ 1 to 13 below are such equations, by means of which resistance may be calculated for any speed and for various car weights. In the formulæ, R is the resistance expressed in pounds per ton, S is the speed expressed in miles per hour, and W is the average weight of the cars in the train expressed in tons. The formulæ are purely empirical, and are simply equations of parabolas so selected as to correspond very closely with the curves of Fig. 11. The correspondence between the formulæ and

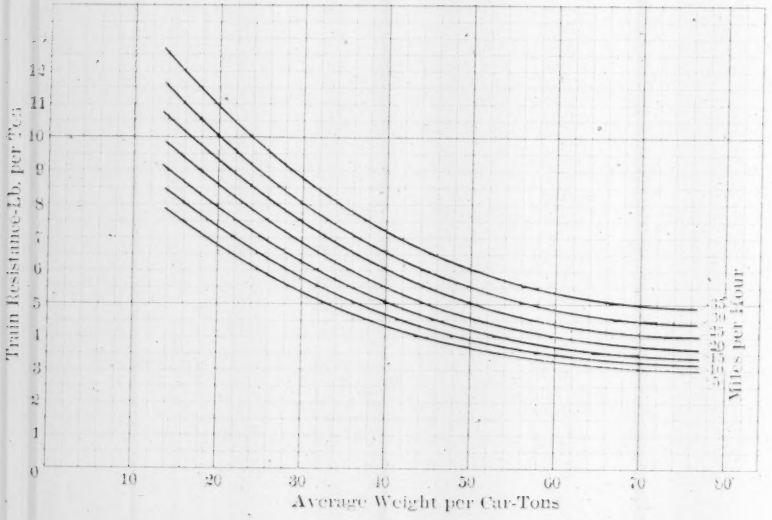

FIG. 10 RELATION BETWEEN RESISTANCE AND AVERAGE CAR WEIGHT AT VARIOUS SPEEDS

the curves is such that the maximum difference between any value of resistance obtained by the formulæ, and the corresponding value obtained from the curves of Fig. 11, is $\frac{1}{2}$ of one per cent. Since these are empirical equations their use should not be extended beyond the speed limits shown on Fig. 11.

74 It is possible to express approximately the facts presented in Fig. 11, in a single equation which includes only the first power of the three variables, R, S and W. Such an equation would obviously be more convenient than the group given below. Several such equations have been derived, each of which well represents, in general,

the results of the tests. Each of them, however, at some points in its range of application, leads to errors as great as 10 per cent. It has been deemed inadvisable to publish a formula containing so great an initial error.

75 *Final Results.* The final results of the research are presented in Fig. 11, in Table 3, and in Formulæ 1 to 13. It is believed that by means of either the figure, the table, or the formulæ, the resistance of ordinary freight trains may be fairly accurately predicted; provided the conditions surrounding their operation are similar to those which prevailed during these tests. These conditions have been fully stated and are restated in the conclusions. It is sufficient to repeat at this point that the results apply to trains running at uniform speed, on tangent and level track of good construction, during weather when the temperature is not lower than 30 deg. fahr. and when the wind velocity does not exceed about 20 miles per hour.

TRAIN RESISTANCE FORMULÆ

When $W = 15$ tons; $R = 7.15 + 0.085\,S + 0.00175\,S^2$ . . (1)
" $W = 20$ " $R = 6.30 + 0.087\,S + 0.00126\,S^2$ . . (2)
" $W = 25$ " $R = 5.60 + 0.077\,S + 0.00116\,S^2$ . . (3)
" $W = 30$ " $R = 5.02 + 0.066\,S + 0.00116\,S^2$ . . (4)
" $W = 35$ " $R = 4.49 + 0.060\,S + 0.00108\,S^2$ . . (5)
" $W = 40$ " $R = 4.15 + 0.041\,S + 0.00134\,S^2$ . . (6)
" $W = 45$ " $R = 3.82 + 0.031\,S + 0.00140\,S^2$ . . ( )
" $W = 50$ " $R = 3.56 + 0.024\,S + 0.00140\,S^2$ . . ( )
" $W = 55$ " $R = 3.38 + 0.016\,S + 0.00142\,S^2$ . . (9)
" $W = 60$ " $R = 3.19 + 0.016\,S + 0.00132\,S^2$ . . (10)
" $W = 65$ " $R = 3.06 + 0.014\,S + 0.00130\,S^2$ . . (11)
" $W = 70$ " $R = 2.92 + 0.021\,S + 0.00111\,S^2$ . . (12)
" $W = 75$ " $R = 2.87 + 0.019\,S + 0.00113\,S^2$ . . (13)

DISCUSSION OF THE RESULTS

76 *Variations in Resistance of Different Trains.* Reference has previously been made to the variations among the points of Figs. 3 to 9. In each figure about one-half of the points lie above the curve there drawn, and their resistance values vary from those of the curve by different amounts. It should be borne in mind that, in these figures each point represents the average resistance which prevailed throughout a particular test, and differences among the points represent, therefore, differences in the mean resistance of the different trains.

77 Among those trains which are regarded as normal there are two or three whose resistance at some speed varies from the mean, as expressed in the curves, by as much as 23 per cent. The great major-

ity, however, vary from this mean by about 10 per cent or less. In Fig. 4, for example, there are nineteen points which lie above the curve, among which the maximum deviation from the mean is 23 per

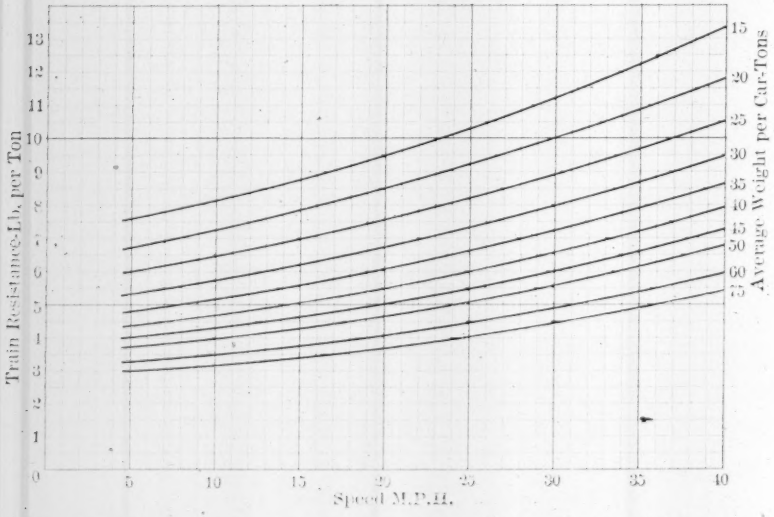

FIG. 11 RELATION BETWEEN RESISTANCE AND SPEED FOR VARIOUS AVERAGE WEIGHTS PER CAR

cent, while the average of the deviation for all nineteen points is 8 per cent. The following table presents similar average deviations above and below the mean for each of Figs. 3 to 9.

AVERAGE DEVIATION OF ALL POINTS IN FIGS. 3 TO 9, FROM THE MEAN AS SHOWN BY THE CURVES THERE DRAWN, EXPRESSED AS PERCENTAGES OF THE CURVE ORDINATES.

	Fig. 3 5 m.p.h.	Fig. 4 10 m.p.h.	Fig. 5 15 m.p.h.	Fig. 6 20 m.p.h.	Fig. 7 25 m.p.h.	Fig. 8 30 m.p.h.	Fig. 9 35 m.p.h.
Points above the curve	11	8	8	11	13	8	7
Points below the curve	13	10	9	8	9	9	9

78 The data present no satisfactory general explanation for these differences in the resistance of different trains of like average weight per car. They may be due to difference in external conditions, or in train condition and make-up. Whatever the explanation it is

TABLE 3 RESISTANCE AT DIFFERENT SPEEDS AND FOR TRAINS OF VARIOUS AVERAGE CAR WEIGHTS

THE VALUES ARE DERIVED DIRECTLY FROM THE CURVES OF FIG. II AND REPRESENT THE FINAL RESULTS OF THE TESTS

Speed Miles per Hour	TRAIN RESISTANCE—POUNDS PER TON — COLUMN HEADINGS INDICATE THE AVERAGE CAR WEIGHTS													Speed Miles per Hour
	15 tons	20 tons	25 tons	30 tons	35 tons	40 tons	45 tons	50 tons	55 tons	60 tons	65 tons	70 tons	75 tons	
5	7.6	6.8	6.0	5.4	4.8	4.4	4.0	3.7	3.5	3.3	3.2	3.1	3.0	5
6	7.7	6.9	6.1	5.5	4.9	4.4	4.1	3.8	3.5	3.3	3.2	3.1	3.0	6
7	7.8	7.0	6.2	5.5	5.0	4.5	4.1	3.8	3.6	3.4	3.2	3.1	3.1	7
8	8.0	7.1	6.3	5.6	5.0	4.6	4.2	3.9	3.6	3.4	3.3	3.2	3.1	8
9	8.1	7.2	6.4	5.7	5.1	4.6	4.2	3.9	3.6	3.4	3.3	3.2	3.1	9
10	8.2	7.3	6.5	5.8	5.2	4.7	4.3	4.0	3.7	3.5	3.3	3.2	3.2	10
11	8.3	7.4	6.6	5.9	5.3	4.8	4.3	4.0	3.7	3.5	3.4	3.3	3.2	11
12	8.4	7.5	6.7	6.0	5.4	4.8	4.4	4.0	3.8	3.6	3.4	3.3	3.3	12
13	8.6	7.6	6.8	6.1	5.5	4.9	4.5	4.1	3.8	3.6	3.5	3.4	3.3	13
14	8.7	7.8	6.9	6.2	5.5	5.0	4.5	4.2	3.9	3.7	3.5	3.4	3.4	14
15	8.8	7.9	7.0	6.3	5.6	5.1	4.6	4.2	3.9	3.7	3.6	3.5	3.4	15
16	9.0	8.0	7.1	6.4	5.7	5.1	4.7	4.3	4.0	3.8	3.6	3.5	3.5	16
17	9.1	8.1	7.2	6.5	5.8	5.2	4.8	4.4	4.1	3.9	3.7	3.6	3.5	17
18	9.3	8.3	7.4	6.6	5.9	5.3	4.8	4.5	4.1	3.9	3.7	3.7	3.6	18
19	9.4	8.4	7.5	6.7	6.0	5.4	4.9	4.5	4.2	4.0	3.8	3.7	3.6	19
20	9.6	8.5	7.6	6.8	6.1	5.5	5.0	4.6	4.3	4.0	3.9	3.8	3.7	20
21	9.7	8.7	7.7	6.9	6.2	5.6	5.1	4.7	4.3	4.1	3.9	3.9	3.8	21
22	9.9	8.8	7.9	7.0	6.3	5.7	5.2	4.8	4.4	4.2	4.0	3.9	3.8	22
23	10.0	9.0	8.0	7.1	6.4	5.8	5.3	4.9	4.5	4.3	4.1	4.0	3.9	23
24	10.2	9.1	8.1	7.3	6.6	5.9	5.4	4.9	4.6	4.3	4.2	4.1	4.0	24
25	10.4	9.3	8.3	7.4	6.7	6.0	5.5	5.0	4.7	4.4	4.2	4.1	4.0	25
26	10.5	9.4	8.4	7.5	6.8	6.1	5.6	5.1	4.8	4.5	4.3	4.2	4.1	26
27	10.7	9.6	8.5	7.7	6.9	6.2	5.7	5.2	4.8	4.6	4.4	4.3	4.2	27
28	10.9	9.7	8.7	7.8	7.0	6.3	5.8	5.3	4.9	4.7	4.5	4.4	4.3	28
29	11.1	9.9	8.8	7.9	7.1	6.5	5.9	5.4	5.0	4.8	4.6	4.5	4.4	29
30	11.3	10.0	9.0	8.0	7.3	6.6	6.0	5.5	5.1	4.9	4.7	4.5	4.5	30
31	11.4	10.2	9.1	8.2	7.4	6.7	6.1	5.6	5.2	5.0	4.8	4.6	4.5	31
32	11.6	10.4	9.3	8.3	7.5	6.8	6.2	5.8	5.3	5.0	4.9	4.7	4.6	32
33	11.8	10.5	9.4	8.5	7.6	7.0	6.3	5.9	5.4	5.2	5.0	4.8	4.7	33
34	12.0	10.7	9.6	8.6	7.8	7.1	6.5	6.0	5.5	5.3	5.1	4.9	4.8	34
35	12.3	10.9	9.7	8.8	7.9	7.2	6.6	6.1	5.7	5.4	5.2	5.0	4.9	35
36	12.5	11.1	9.9	8.9	8.0	7.4	6.7	6.2	5.8	5.5	5.3	5.1	5.0	36
37	12.7	11.2	10.0	9.0	8.2	7.5	6.9	6.4	5.9	5.6	5.4	5.2	5.1	37
38	12.9	11.4	10.2	9.2	8.3	7.6	7.0	6.5	6.0	5.7	5.5	5.3	5.2	38
39	13.1	11.6	10.4	9.4	8.5	7.8	7.1	6.6	6.2	5.8	5.6	5.4	5.3	39
40	13.4	11.8	10.6	9.5	8.6	7.9	7.3	6.8	6.3	6.0	5.7	5.6	5.5	40

significant that about one-half of the trains experimented upon developed a resistance about 9 per cent in excess of the mean resistance which would be predicted by the use of Figs. 3 to 9 and Figs. 10 and 11. Obviously a similar excess may be expected with any train; it is suggested therefore that in determining the resistance of trains on *level tangent track* for the purpose of rating locomotives under operating conditions which demand conservative ratings, 9 per cent be added to the resistance values obtained from the curves, tables and equations presented. Such considerations are of little practical importance in rating locomotives for speeds above 15 miles per hour. In such cases an excess in resistance over that expected can result in nothing more serious than failure to realize the expected train speed.

79 It should be understood that this 9 per cent allowance is intended to cover probable variations in the resistance of different trains under normal operating conditions. It in no way takes the place of that additional reserve which must be allowed to cover unusual variations in resistance due to low temperatures or high winds, nor of that reserve in the tractive effort of the locomotive necessitated by operating conditions which reduce the efficiency of the locomotive itself.

80 *Tests which Present Abnormal Resistance Values.* There are four points in Figs. 3 to 9 whose deviation from the curves is so great as to demand special examination. These are the points corresponding to Tests S-1034, S-1074, S-1080, and S-1031 (points 34, 74, 80, and 31). These tests show a persistent and great variation from the mean at various speeds. The trains of Tests 1034, 1074, and 1080 were like in average car weights, less than 23 tons, and in containing a large proportion of empty gondolas— 99, 98, and 84 per cent, respectively. Any explanation based on train composition is however nullified by the fact that the trains of Tests 1016, 1043, and 1063, which show close correspondence with the curves, had similar average car weights and contained almost equally large proportions of empty gondolas. Weather and wind conditions likewise offer no explanation of the divergences presented by these three tests. Explanations are rendered more difficult by the fact that while the trains of Tests 1034 and 1074 show unusually high resistance, the resistance in Test 1080 is exceptionally low. The abnormalities presented by these three trains have therefore been accepted as unexplained by the data at hand.

81 The resistance of the train of the fourth test mentioned above (S-1031) was low at all speeds. This train had an average car weight

of 20.7 tons, contained 94 per cent of box cars, and was only 1425 ft. long. Other test trains of similar average car weight differ generally in having less than 60 per cent of box cars, and all in being 2400 ft. or more in length. Taking into consideration all the data, neither fact seems to offer an adequate explanation, however, of the variations exhibited by this train.

82 *Car Weight as a Basis of Expression.* Objection may be made to the form of expression adopted in Figs. 3 to 9 and in Fig. 10, in which the resistance is expressed solely in terms of average car weight, to the apparent neglect of the influence of those elements of resistance, such as air resistance, which are independent of weight, and which probably vary only with the number of cars in the train. The neglect is only apparent, however, for the process by which Fig. 10 was derived involves, although indirectly, recognition of the influence of the number of cars. It is quite likely that if Fig. 10 were applied to determine the total resistance of a single car, the result would be in error.

83 Whatever objection may be urged against the form of expression adopted, it remains true that Fig. 10 rests upon experimental results obtained with trains of usual length and that in practice one is not likely to encounter trains which present in this respect any extreme variation from the test data. The form of expression will not lead to error unless misapplied, and it was chosen because the results may be conveniently used in establishing tonnage ratings.

84 It might likewise have been more rational to express the resistance in terms of load per axle instead of load per car, since the latter can operate to cause variations in resistance only in so far as it affects the former. Since, however, all American freight cars have four axles, the expression in either form would be identical. Convenience in application warrants the choice made in this respect also.

85 *Effect of Variety in Car Weight upon Total Train Resistance.* In Fig. 10 those portions of the curves which apply to average car weights below 20 tons were derived from trains which were quite homogeneous in their make-up as regards weight per car. These trains were necessarily composed almost exclusively of empty cars, since an average car weight of 20 tons or less cannot otherwise be obtained with cars of current design; and being empty they will be uniform in weight. Similarly for average car weights above 55 to 60 tons the test trains were necessarily uniform in make-up. For trains of average car weights below 20 and above 60 tons the curves of Fig. 10 are accepted, therefore, as valid, and applicable to any train to be met with in practice.

86 In Fig. 10, those portions of the curves corresponding to car weights of from 20 to 60 tons were, on the other hand, derived from trains which presented considerable diversity in make-up as regards weight per car. Some of these trains were composed almost entirely of loaded cars, others contained large proportions of both empty and loaded cars. In presenting the results in the form adopted in Fig. 10 (and Fig. 11), the assumption is that the curves there drawn will be used throughout their entire range of average car weight to determine the total resistance of both homogeneous and mixed trains, and that, when so applied, they will lead to no material error. In view of the facts just stated it is pertinent to inquire whether this assumption is justifiable.

87 Assume two trains of equal tonnage, and of the same average weight per car. Assume further that one is composed of cars uniform in weight, and the other of cars of different individual weights. Now if such trains are to have equal total resistance, it can be shown that the variation in resistance per car of the individual cars must be directly proportional to their weight. This implies that the curve showing the relation between total car resistance and car weight at a given speed must be a straight line, if homogeneous and mixed trains are to have equal total resistances at this speed. From Fig. 10 there have been derived such curves, showing the relation between car resistance and car weight. These curves (not shown in the report) correspond quite closely, but not exactly, with straight lines; and the correspondence is especially close for those portions of the curves which apply to car weights between 20 and 60 tons. From these facts we may conclude that the curves of Fig. 10 are not quite equally applicable to mixed and homogeneous trains, but nearly so, and that if the curves are applied to both kinds of trains, we may expect a slight error in the resulting total train resistance. The amount of such error is indicated by the following examination of a specific case.

88 Assume two trains, *A* and *B*, the first homogeneous, the second mixed, as regards car weight. Train *A* is composed of 60 cars each weighing 45 tons, and its total weight is 2700 tons. Train *B* is composed of 30 cars of 70 tons each and 30 cars of 20 tons each; its total weight is 2700 tons and its average car weight is 45 tons. Train *B* presents about as great a diversity in car weight as may be encountered in current practice. Both trains have equal tonnage and the same average weight per car. Assume that the total resistance of these two trains at a speed of 5 miles per hour is to be determined. By the procedure which it is intended shall usually be followed in using

Fig. 10, the resistance for an average car weight of 45 tons, at 5 miles per hour, is found to be 4.0 lb. per ton; and the total resistance of either train *A* or train *B* is $2700 \times 4.0 = 10{,}800$ lb.

89 Train *B*, however, may be considered as made up of two shorter homogeneous trains of average car weights of 20 and 70 tons respectively, and the resistance of each may be determined from those portions of the curves of Fig. 10 about whose validity no question is raised. From Fig. 10, the resistance at 5 miles per hour for a car weight of 20 tons is found to 6.8 lb. per ton, and for a car weight of 70 tons, 3.1 lb. per ton. By the use, therefore, of these portions of the curves of Fig. 10, the total resistance of train *B* is found to be $30 \times 20 \times 6.8 + 30 \times 70 \times 3.1 = 10{,}590$ lb., which differs from the resistance previously found by 2 per cent. If similar analysis be made for a speed of 40 miles per hour, the corresponding difference is found to be 4 per cent. If these differences be accepted as a measure of the maximum error likely to result from the indiscriminate application of the curves of Fig. 10 to mixed and homogeneous trains, we may conclude that for purposes of rating locomotives the results of the tests as expressed in Fig. 10 and 11 and Table 3 may be so applied without material error.

90 *The Influence of Speed on Resistance.* Within the last two years the opinion has been expressed in some quarters that train resistance between speeds of 5 and 35 miles per hour is constant. It is proper to point out that there is nothing in the data here presented to support such a conclusion.

91 *The Influence of Wind Velocity on Resistance.* The wind velocities prevailing during the tests were generally less than 20 miles per hour. The data do not permit the influence of such winds to be differentiated from the other elements affecting resistance; but they do warrant the conclusion that this influence is small. In the introduction, train resistance was defined as the resistance in still air, whereas throughout the report the term is applied to the test results from which the influence of wind has not been eliminated. This inconsistency has been deliberately incurred to avoid unwieldy expression, and is partially justified by the facts just stated.

92 *Comparison with other Experiments.* There is no point in comparing the results of these tests with formulæ in which the influence of car weight is given no consideration, nor with those not derived from tests on American cars of recent design. The results obtained on the Chicago, Burlington and Quincy Railroad and on the Pennsyl-

vania Railroad, and recently published by Mr. F. J. Cole are selected for comparison.

93 The results obtained on the Chicago, Burlington and Quincy road ("curve No. 1, for temperatures above 30 deg. fahr. and no wind") apply to a speed of 20 miles per hour. Compared with the curve for 20 miles per hour in Fig. 10, they show resistance values from 35 to 60 per cent lower than the corresponding results of these tests. The Pennsylvania Railroad results are supposed to be equally applicable at all speeds between 5 and 30 miles. When plotted on Fig. 10 of this report they show very close correspondence with the curve there drawn for 10 miles per hour, for car weights from 25 to 70 tons; while for car weights below 25 tons they indicate resistance values as much as 20 per cent in excess of the results obtained during these tests.

[1] Railway Age Gazette, August 27 to October 1, 1909.

APPENDIX

RAILWAY TEST CAR NUMBER 17

The dynamometer car by means of which these tests were made was built in 1900. Under the arrangements then entered into, the car itself was built and has been since maintained by the Illinois Central Railroad; while the University has supplied all apparatus, and has manned and operated the car. Both the car body and the apparatus were remodeled in 1907.[1]

2 The car body was especially designed for its purpose. It is 40 ft. long over the end-sills, and 8 ft. 4 in. wide inside. The central sills and the platforms are of steel, while the remainder of the construction is of wood. The general design of the car is shown in Fig. 1, and an interior view is shown in Fig. 2. The working space occupies about two-thirds of the length of the car, and in it are placed the recording apparatus, the auxiliary instruments, the storage batteries, the work bench, etc.

3 During the tests which are here reported the test-car apparatus made continuous autographic records of drawbar pull, speed, time, mile-post positions, air-brake cylinder pressure, wind velocity with respect to the car, and wind direction with respect to the longitudinal axis of the car. These records were made upon a chart 36 in. wide, drawn across the table of the recording apparatus. During these tests this chart was driven by gearing from the axle of the central truck below the car, so that its travel was proportional to the travel of the car itself. A view of the recording apparatus is shown in Fig. 3.

4 Fig. 4 is reproduced from a tracing of a portion of the chart made during test S–1057 of this series. The only lines there shown which do not appear on the original record, are the profile and the transverse lines which mark the limits of one of the sections selected for calculation. These lines and some of the explanatory lettering have been added to the tracing, in order to make clearer the significance of the various records.

5 The total pull which comes upon the measuring drawbar of the car is transmitted to oil contained in the receiving cylinder, whose design is shown in Fig. 5. This cylinder is hung from the center sills, immediately behind the drawbar yoke. Its inside diameter is 10 in., and its piston is 7 1/2 in. long. Both cylinder and piston are carefully ground to an exact fit and no piston packing is used. The pull is transmitted from the drawbar yoke to the piston, through a roller-borne yoke; and the whole device is practically frictionless. Such leakage of oil as takes place proceeds so slowly as to prove of no inconvenience, even when operating under maximum pull. The cylinder may be refilled with oil

[1] A more detailed description of the present equipment is contained in an article by F. W. Marquis, which appeared in the Railway Age Gazette of February 19, 1909.

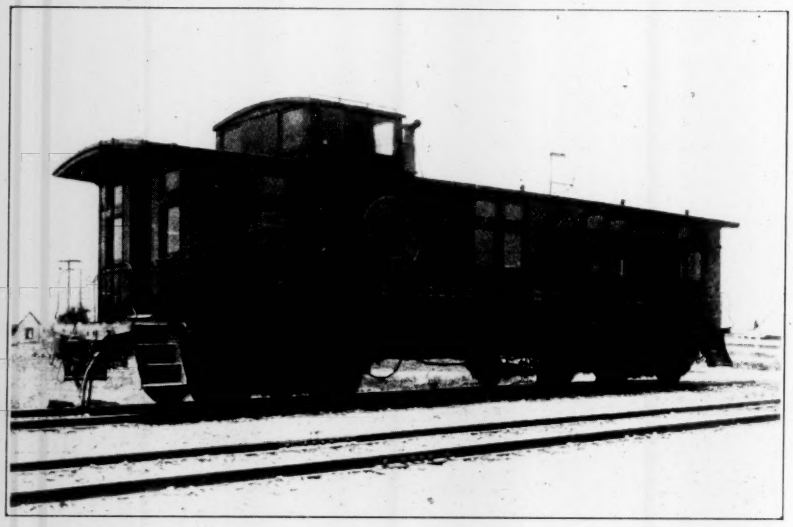

FIG. 1 RAILWAY TEST CAR NO. 17

FIG. 2 INTERIOR OF TEST CAR NO. 17

by means of a pump within the car while the car is in operation, without impairing the accuracy of the record. The pressure of the oil in this receiving cylinder is transmitted to the cylinder of an indicator located upon the table within the car. This indicator is identical in design with one of the modern types of steam-engine indicators; although it is larger and heavier throughout. During its ten years of service this dynamometer has demonstrated its reliability and accuracy.

6 Two speed records are shown on the chart, and both are used. The one is obtained from a speed recorder which resembles in design a "flyball" engine governor. This instrument is used in measuring speeds above 15 miles per hour. The second record is obtained from a chain-driven Boyer speed recorder, geared to run at about three times its usual speed. This record is used for speeds up to 35 miles per hour. Within their respective ranges, both instruments produce accurate speed curves.

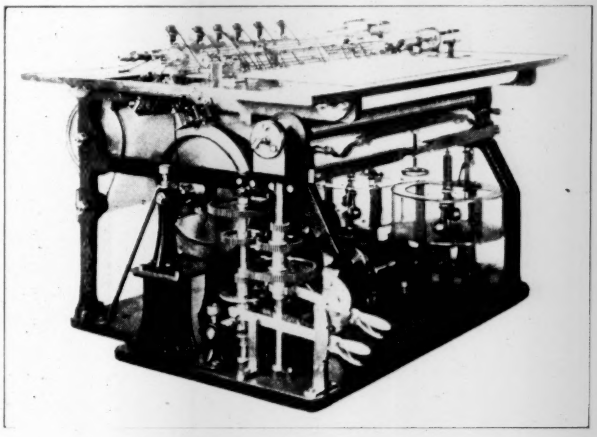

FIG. 3 THE RECORDING APPARATUS

7 The air-brake cylinder of the test car is connected to the cylinder of an ordinary steam engine indicator, which is mounted upon the table and which draws a curve of air-brake cylinder pressure.

8 The velocity of the wind with respect to the car is obtained by means of a Robinson cup-anemometer of the standard United States Weather Bureau type, which is so mounted that the cups revolve 32 in. above the car roof. This instrument controls an electric circuit, which operates an electro-magnet connected to the recording pen. By means of this magnet, offsets are made in the line drawn by the pen. During the time which elapses between two succeeding offsets, the actual movement of the wind amounts to 0.2 of a mile.

9 The direction of the wind with respect to the longitudinal axis of the car, is derived from a wind vane mounted 3 ft. above the car roof. The spindle of the vane extends downward to a point above the recording apparatus, and terminates there in a crank which is parallel to the vane. This crank is con-

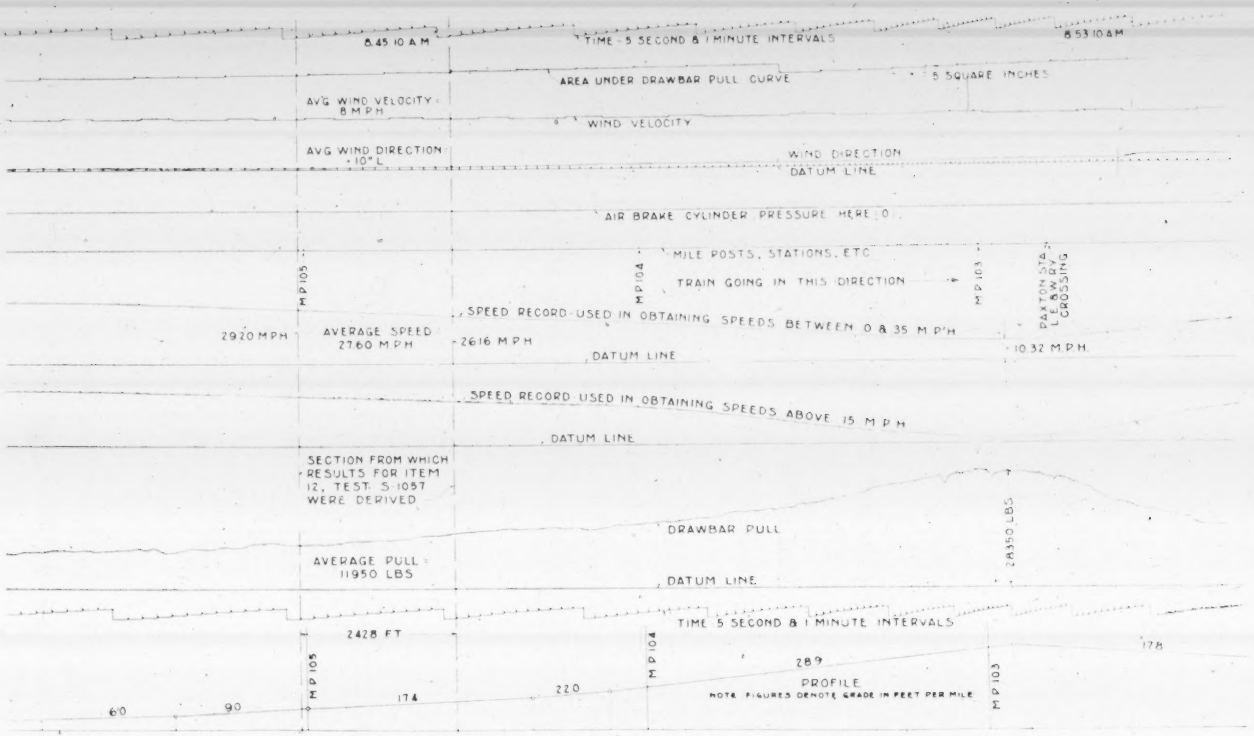

FIG. 4 A PORTION OF THE CHART FROM TEST S-1057

nected to the recording pen through a rod with a yoke end. The ordinate of the curve drawn by this pen is proportional to the sine of the angle made by the vane with the car axis. The offsets in the datum line for this curve which appear in Fig. 4, indicate that the vane, at the moment, was pointed toward the front end of the car.

10 Fig. 4 shows a record of "area under the curve of pull" which is made by means of a recording planimeter mounted on the table. This record is inaccurate and was not used in these calculations.

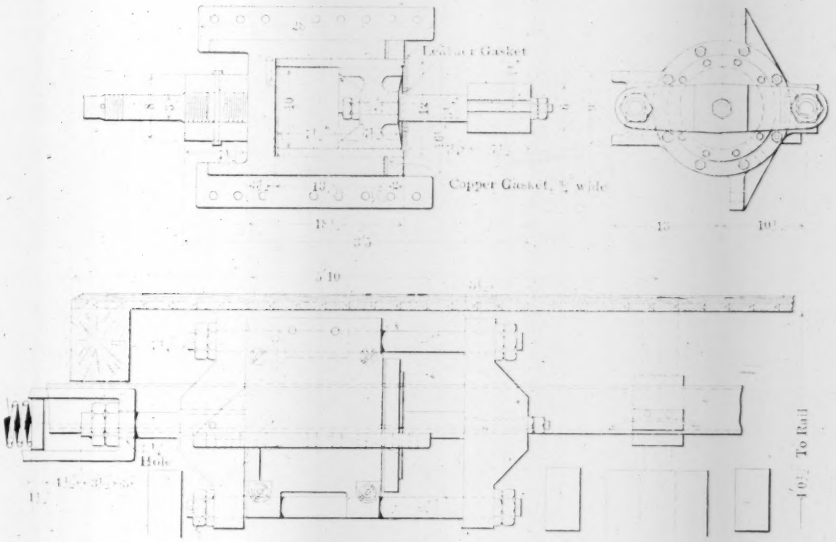

FIG. 5 RECEIVING CYLINDER OF THE DYNAMOMETER

EXACT COORDINATES FOR THE CURVES OF FIGS. 10 AND 11.

11 The original drawings from which Figs. 10 and 11 of the paper have been reproduced, were drawn to a scale about four times as great as that of the cuts. From these original drawings, the values of the coordinates of the various curves of both figures have been determined as accurately as possible; and these values are presented in Tables 1 and 2 herewith.

12 The curves of Fig. 10 (and of Figs. 3 to 9) of the paper may be accurately reproduced by the use of Table 4; and the curves of Fig. 11 may be reproduced from the values given in Table 2 herewith. The tables are presented merely to permit the accurate reproduction, to any scale, of the curves of the report; and are not intended for use in determining values of resistance. For the latter purpose Table 3 of the paper is more convenient and sufficiently accurate.

TABLE 1*a* RESISTANCE AT DIFFERENT SPEEDS AND FOR TRAINS OF VARIOUS AVERAGE CAR WEIGHTS

THIS TABLE PRESENTS THE COÖRDINATES OF THE ORIGINAL CURVES FROM WHICH FIGS. 3 TO 9 AND FIG. 10 WERE REPRODUCED

Average Weight per Car Tons		Train Resistance—Pounds per Ton — column headings indicate the various speeds							Average Weight per Car Tons	
		5 m.p.h.	10 m.p.h.	15 m.p.h.	20 m.p.h.	25 m.p.h.	30 m.p.h.	35 m.p.h.		
15		7.62	8.20	8.81	9.36	10.37	11.24	12.25		15
	16	7.44	8.00	8.61	9.34	10.13	10.98	11.95	16	
	18	7.10	7.63	8.22	8.92	9.68	10.47	11.39	18	
	20	6.77	7.30	7.85	8.53	9.26	10.00	10.89	20	
	22	6.45	6.97	7.49	8.16	8.84	9.56	10.41	22	
	24	6.16	6.64	7.14	7.79	8.46	9.16	9.94	24	
25		6.02	6.50	6.98	7.62	8.28	8.95	9.72		25
	26	5.88	6.35	6.81	7.44	8.10	8.77	9.52	26	
	28	5.61	6.07	6.51	7.11	7.76	8.40	9.12	28	
	30	5.38	5.80	6.23	6.80	7.43	8.05	8.75	30	
	32	5.13	5.54	5.98	6.51	7.12	7.72	8.40	32	
	34	4.92	5.31	5.72	6.24	6.82	7.40	8.06	34	
35		4.82	5.20	5.61	6.11	6.68	7.26	7.91		35
	36	4.72	5.10	5.50	5.99	6.55	7.11	7.77	36	
	38	4.55	4.90	5.28	5.74	6.29	6.83	7.48	38	
	40	4.38	4.70	5.06	5.50	6.03	6.57	7.20	40	
	42	4.22	4.52	4.88	5.29	5.80	6.32	6.95	42	
	44	4.08	4.38	4.70	5.09	5.59	6.10	6.71	44	
45		4.01	4.30	4.61	4.99	5.49	6.00	6.60		45
	46	3.95	4.21	4.52	4.90	5.38	5.90	6.49	46	
	48	3.82	4.08	4.38	4.71	5.20	5.71	6.28	48	
	50	3.72	3.96	4.24	4.56	5.03	5.52	6.10	50	
	52	3.61	3.85	4.11	4.42	4.88	5.36	5.91	52	
	54	3.52	3.75	3.99	4.30	4.74	5.20	5.74	54	
55		3.48	3.71	3.94	4.25	4.68	5.12	5.67		55
	56	3.43	3.67	3.90	4.20	4.62	5.05	5.60	56	
	58	3.37	3.58	3.81	4.10	4.50	4.93	5.47	58	
	60	3.30	3.50	3.73	4.02	4.42	4.83	5.36	60	
	62	3.23	3.44	3.67	3.97	4.34	4.74	5.27	62	
	64	3.18	3.39	3.60	3.90	4.29	4.68	5.18	64	
65		3.15	3.36	3.58	3.88	4.25	4.64	5.14		65
	66	3.12	3.32	3.55	3.85	4.22	4.61	5.11	66	
	68	3.09	3.30	3.50	3.80	4.18	4.57	5.06	68	
	70	3.05	3.26	3.47	3.76	4.13	4.52	5.01	70	
	72	3.02	3.22	3.44	3.73	4.10	4.49	4.98	72	
	74	3.01	3.19	3.42	3.71	4.08	4.48	4.93	74	
75		3.00	3.18	3.41	3.70	4.07	4.47	4.91		75

TABLE 2*b* RESISTANCE AT DIFFERENT SPEEDS AND FOR TRAINS OF VARIOUS AVERAGE CAR WEIGHTS

THIS TABLE PRESENTS THE COÖRDINATES OF THE ORIGINAL CURVES FROM WHICH FIG. 11 WAS REPRODUCED

Speed Miles per Hour	Train Resistance—Pounds per Ton — Column headings indicate the average car weights													Speed Miles per Hour
	15 tons	20 tons	25 tons	30 tons	35 tons	40 tons	45 tons	50 tons	55 tons	60 tons	65 tons	70 tons	75 tons	
5	7.62	6.77	6.02	5.38	4.82	4.39	4.01	3.72	3.49	3.30	3.16	3.05	3.00	5
6	7.73	6.86	6.12	5.46	4.90	4.43	4.07	3.77	3.52	3.33	3.19	3.08	3.03	6
7	7.83	6.97	6.21	5.53	4.98	4.50	4.12	3.81	3.56	3.37	3.23	3.12	3.07	7
8	7.96	7.06	6.31	5.62	5.04	4.57	4.18	3.86	3.60	3.40	3.26	3.16	3.10	8
9	8.07	7.18	6.40	5.71	5.11	4.62	4.22	3.90	3.64	3.44	3.30	3.20	3.13	9
10	8.19	7.29	6.50	5.80	5.20	4.69	4.28	3.96	3.69	3.49	3.34	3.24	3.18	10
11	8.30	7.40	6.60	5.90	5.29	4.76	4.33	4.00	3.73	3.52	3.38	3.29	3.21	11
12	8.42	7.51	6.71	5.98	5.37	4.83	4.40	4.04	3.78	3.58	3.42	3.33	3.26	12
13	8.56	7.63	6.81	6.08	5.46	4.90	4.47	4.11	3.83	3.62	3.47	3.38	3.31	13
14	8.70	7.76	6.92	6.18	5.53	4.98	4.53	4.18	3.89	3.68	3.52	3.43	3.36	14
15	8.82	7.88	7.01	6.28	5.64	5.06	4.60	4.24	3.94	3.73	3.57	3.48	3.41	15
16	8.98	8.00	7.12	6.39	5.73	5.13	4.68	4.31	4.00	3.80	3.62	3.53	3.47	16
17	9.10	8.13	7.24	6.49	5.82	5.23	4.75	4.38	4.05	3.86	3.68	3.60	3.52	17
18	9.25	8.27	7.37	6.60	5.92	5.32	4.83	4.45	4.12	3.92	3.74	3.66	3.58	18
19	9.40	8.40	7.49	6.71	6.01	5.41	4.91	4.52	4.19	3.98	3.81	3.72	3.64	19
20	9.56	8.53	7.60	6.82	6.11	5.50	5.00	4.60	4.27	4.04	3.88	3.79	3.71	20
21	9.71	8.69	7.72	6.93	6.22	5.60	5.08	4.69	4.32	4.11	3.94	3.85	3.78	21
22	9.88	8.82	7.86	7.03	6.33	5.70	5.17	4.78	4.41	4.18	4.00	3.92	3.84	22
23	10.02	8.97	7.99	7.14	6.44	5.80	5.27	4.86	4.49	4.25	4.07	3.99	3.92	23
24	10.20	9.11	8.11	7.27	6.55	5.90	5.37	4.94	4.58	4.33	4.15	4.06	3.98	24
25	10.37	9.26	8.25	7.40	6.67	6.01	5.46	5.03	4.66	4.41	4.23	4.13	4.04	25
26	10.52	9.42	8.38	7.52	6.79	6.11	5.57	5.12	4.75	4.50	4.31	4.21	4.12	26
27	10.71	9.57	8.51	7.65	6.91	6.21	5.67	5.22	4.83	4.58	4.40	4.29	4.20	27
28	10.89	9.72	8.67	7.78	7.01	6.33	5.78	5.32	4.92	4.67	4.48	4.38	4.29	28
29	11.06	9.89	8.81	7.91	7.12	6.45	5.88	5.43	5.01	4.76	4.57	4.46	4.36	29
30	11.25	10.03	8.96	8.04	7.26	6.58	5.99	5.53	5.11	4.86	4.66	4.53	4.45	30
31	11.43	10.20	9.10	8.18	7.39	6.71	6.10	5.64	5.21	4.95	4.75	4.63	4.53	31
32	11.63	10.37	9.26	8.31	7.51	6.83	6.21	5.76	5.32	5.04	4.85	4.73	4.62	32
33	11.84	10.53	9.41	8.46	7.63	6.96	6.33	5.87	5.43	5.15	4.95	4.83	4.72	33
34	12.04	10.71	9.57	8.60	7.78	7.08	6.47	5.99	5.54	5.26	5.05	4.92	4.82	34
35	12.25	10.89	9.72	8.75	7.91	7.20	6.60	6.10	5.67	5.36	5.16	5.01	4.92	35
36	12.47	11.07	9.89	8.90	8.04	7.35	6.73	6.23	5.78	5.48	5.27	5.12	5.01	36
37	12.69	11.23	10.04	9.04	8.19	7.49	6.87	6.36	5.90	5.59	5.28	5.22	5.12	37
38	12.91	11.42	10.21	9.20	8.33	7.64	7.00	6.49	6.02	5.71	5.48	5.33	5.22	38
39	13.12	11.61	10.39	9.36	8.48	7.79	7.13	6.63	6.15	5.83	5.60	5.44	5.33	39
40	13.35	11.80	10.55	9.51	8.62	7.93	7.29	6.78	6.28	5.95	5.72	5.55	5.45	40

THE STRENGTH OF PUNCH AND RIVETER FRAMES MADE OF CAST IRON

By Prof. A. Lewis Jenkins, Cincinnati, O.

Member of the Society

The actual stress relations existing in a simple cast-iron beam when subjected to a load have been the object of many investigations conducted by engineers during the past hundred years. These investigations consisted of numerous experiments and mathematical deductions for the determination of the elastic laws and physical constants involved in the analysis of stresses in straight cast-iron beams, but did not include the necessary deductions for the case of curved beams similar to punch and riveter frames.

2 The object of this article is to determine experimentally the relation between the ultimate strength of curved cast-iron specimens similar to punch and shear frames and the ultimate strength of the attached test bars and to compare the experimental results with those determined by the various methods of analysis used in designing castings of similar shape. In making this investigation many publications relating to the subject have been studied and the most important results many of which are only of historical interest are stated and discussed in the appendix.

FACTOR OF SAFETY

3 It has been contended that the ultimate strength of cast-iron machine parts is of no practical value since the working stress seldom exceeds 5000 lb. per square inch, and as Hook's law is sensibly true within this limit the ordinary formulas for beams should apply with a fair degree of accuracy. It is frequently desirable, however, to know the ultimate strength of such elements as punch frames in order to design the safety link which is supposed to break and save the frame in case of an accident or overload.

The American Society of Mechanical Engineers, 29 West 39th Street, New York. All papers are subject to revision.

4 The ordinary formulas assume that the stress in a beam is directly proportional to the load and factors of safety based on the ultimate tensile strength of the material are the same as if based on the ultimate strength of the beam. This is not true, however, for cast-iron because the stress is not directly proportional to the load. A beam 1 in. square and 12 in. between supports made of cast iron having an ultimate tensile strength of 20,000 lb. per square inch would support a load of 222 lb. midway between the supports with a factor of safety of 5 based on the ultimate tensile strength. Such a beam would probably fail, however, under a load in the middle of about 2000 lb. giving a factor of safety of 9 based on the breaking load of the beam. Hence, by using the ordinary beam formulas in designing straight cast-iron machine elements that are to be subjected to bending strains the actual working stress may be close to that desired, but the actual factor of safety based on the strength of the casting will be much greater than the factor of safety based on the tensile strength of the material.

Methods Employed in Making Tests

5 For many years engineers have recognized the value of experimental data on the strength of cast-iron beams and a considerable volume of literature on this subject is now available. There are, however, no published data on the strength of curved beams similar in shape to punch frames. The results of some experiments on the strength of crane hooks published in the October 7, 1909, issue of the American Machinist, seem to show that the Pearson-Andrews formula is true within the elastic limit for steel specimens of small throat depth. It will be noticed, however, that the value used for Poisson's ratio does not compare favorably with the values of this constant usually given for steel.

6 In view of the fact that there seems to be no rational method for determining the strength of curved cast-iron beams the writer believes that some experimental data on this subject may be of practical value and also serve as an incentive for further investigation by others.

7 Three forms of test specimens as shown in Figs. 1, 2 and 3 were used for these tests. These specimens were carefully designed with a view to having them of equal strength throughout, the calculations being based on the ordinary formulas for beams. To prevent any torsional moment due to eccentric loading the specimens were provided with two small hemispherical projections for receiving the load. A round and a rectangular test bar were cast with each specimen as

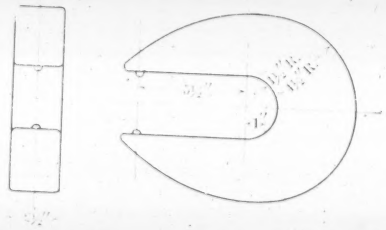

FIG. 1 DIMENSIONS OF SPECIMENS 1, 2 AND 3, PLATE I

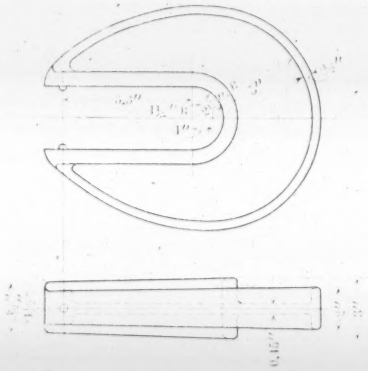

FIG. 2 DIMENSIONS OF SPECIMENS 4, 5, 7 AND 12, PLATES I AND II

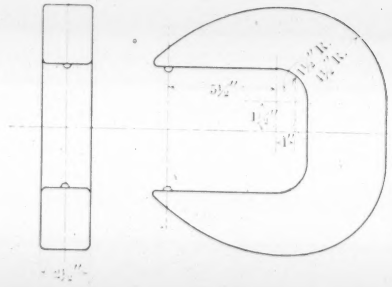

FIG. 3 DIMENSIONS OF SPECIMENS 16 AND 17, PLATE III

FIG. 4 ROUND AND RECTANGULAR TEST BARS CAST WITH EACH SPECIMEN

shown in Fig. 4. The castings were made by the Buckeye Foundry Company of Cincinnati and poured from a heat run to cast lathe beds.

8 The 100,000 lb. Riehle testing machine shown in Fig. 5 was used for making the tests. Fig. 6 shows the method employed for applying the load. Two stirrups forged from 1 in. by 3 in. steel and then tem-

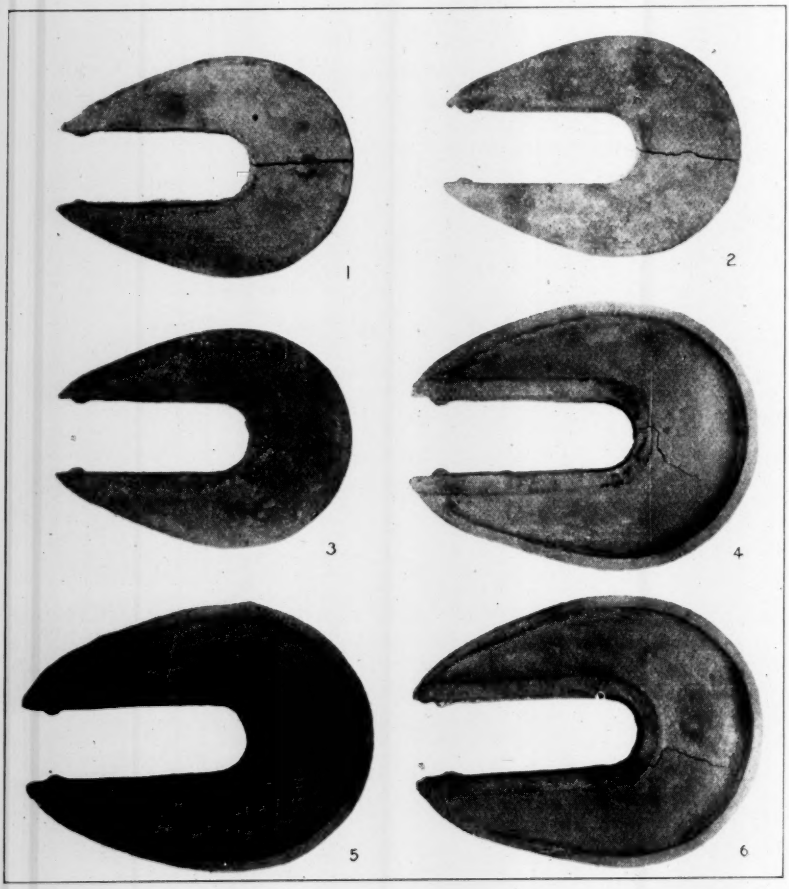

PLATE I CURVED CAST-IRON SPECIMENS BROKEN BY TESTS

pered were held in the grips and received the castings. The tangs of the autographic recorder were placed between the casting and the stirrups, thereby eliminating the deflection of the stirrups and the bearing projections on the castings.

SIZE OF SPECIMENS

9. Plates I, II, and III were made from photographs of the broken specimens. The dimensions of specimens No. 1, 2 and 3 are shown in Fig. 1, and Fig. 2 shows the dimensions of specimens No. 4, 5, 7 and 12. The flanges on No. 12 were partially removed, but this did not

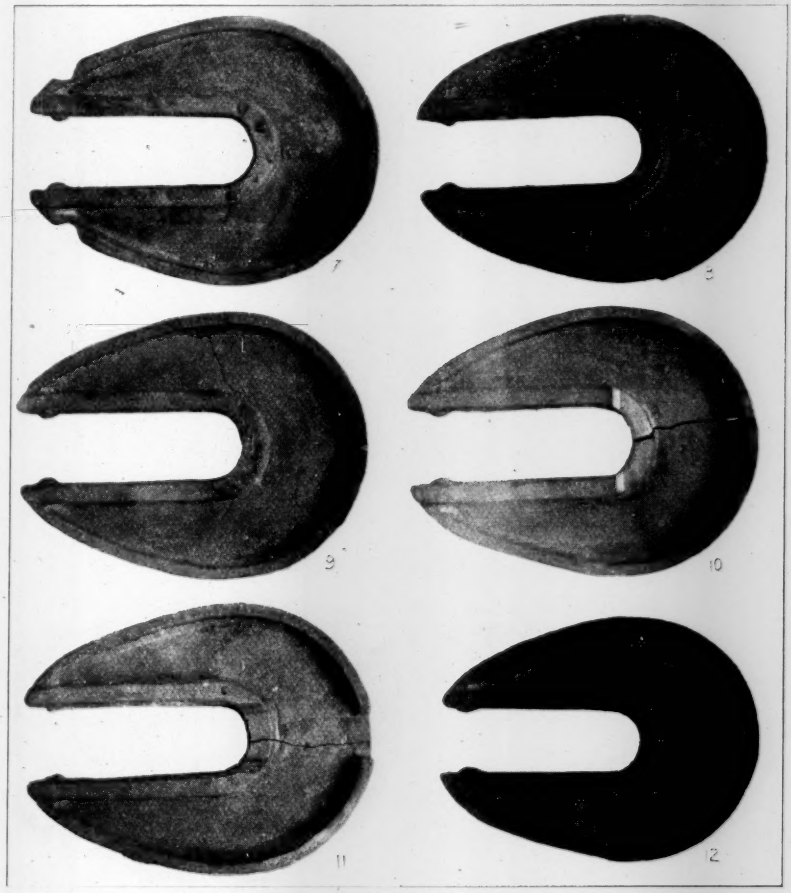

PLATE II. CURVED CAST-IRON SPECIMENS BROKEN BY TESTS

affect its strength. Specimens No. 9, 10, 11 and 13 were cast from the same pattern as No. 4 and the flanges altered as shown. No. 6 was cast from the same pattern as No. 4 after the thickness of the web had been increased to 0.93 in. No. 14 differs from No. 6 in that the fillet behind the flange is larger. No. 15 is the same as No. 14 with the out-

side flange removed. Fig. 3 shows the dimensions of No. 16 and 17. No. 18 is the same as No. 16 with depth of spine reduced to 3.4 in.

RESULTS OF TESTS

10 Table 1 gives the results of the tests. The unit tensile strength of the test bars varied from 18,600 to 24,400 lb., and the flexural

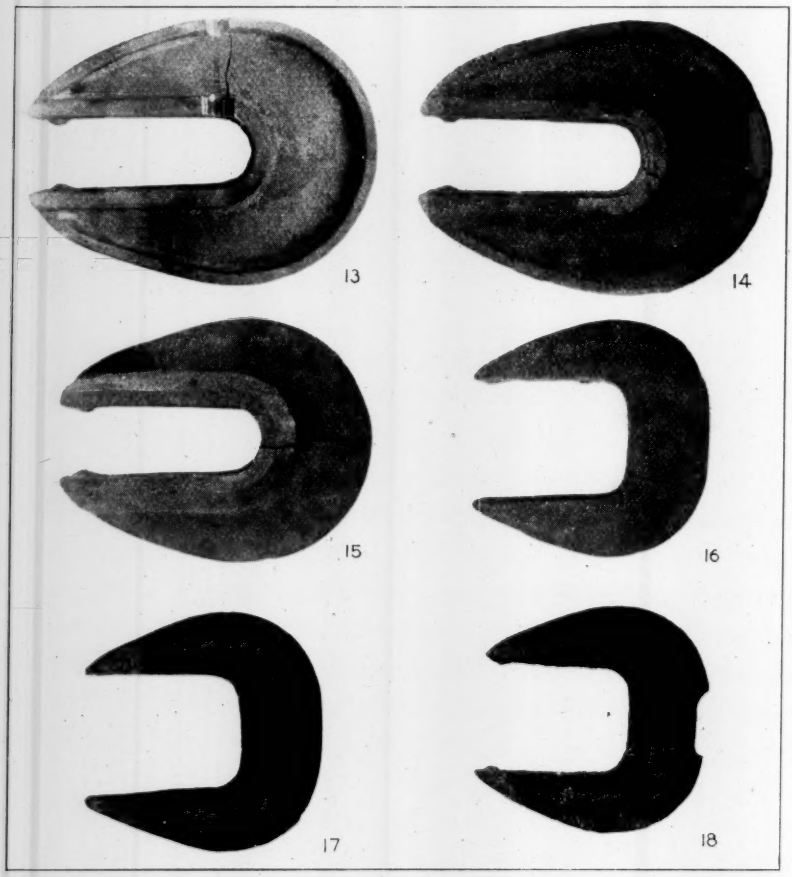

PLATE III CURVED CAST-IRON SPECIMENS BROKEN BY TESTS

strength varied from 36,400 to 46,400 lb. The variations in the values of K seem to indicate that no definite relation exists between the tensile and flexural strength of small test bars.

11 The determination of the correct value for the radius of curvature of the gravity axis is extremely important and considerable care

was exercised in attempting to get accurate measurements of these values by plotting the curve of the gravity axis and finding its radius of curvature at the section considered in each case. It is possible for the personal equation to cause sufficient variations in these values to affect appreciably the final results. The chances for errors in plotting the transformed curves and measuring their areas is also worthy of the designer's consideration in choosing a formula.

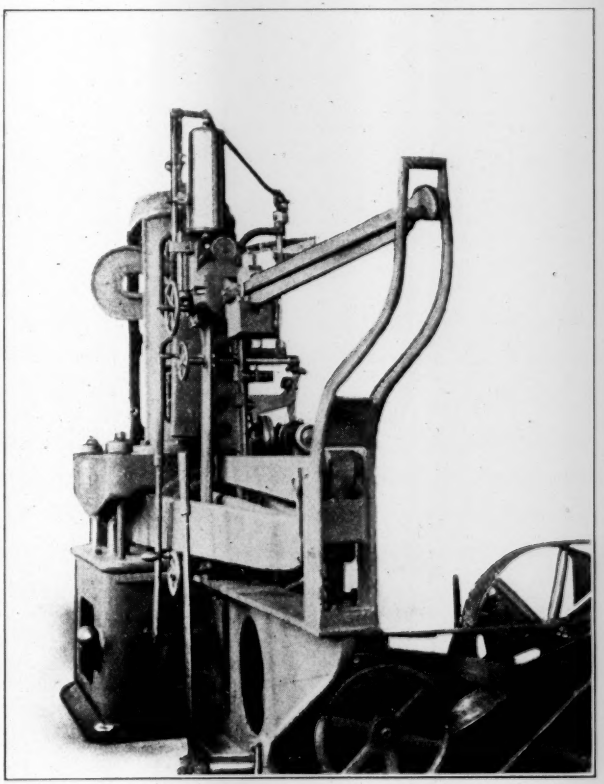

FIG. 5 100,000-LB. RIEHLE TESTING MACHINE

12 The average dimensions of castings No. 1, 2, and 3, Plate I, cast from the same pattern, are given in Fig. 1. Each casting was carefully measured and the values used in the respective calculations. The average tensile strength of the three test bars is 18,907 lb. per square inch. The maximum stress in the castings at failure as given by the beam formula is 14.2 per cent less than the tensile strength of the

test bar, whereas the stress according to the Résal and Pearson-Andrews formulas is 43.4 per cent and 75.4 per cent in excess of the strength of the test bar.

13 From Keep's experiments it is known that the unit strength of cast-iron bars of the same composition decreases as the area increases; hence, it is reasonable to suppose that the unit stress given by the beam formula is very close to the actual ultimate strength of the mate-

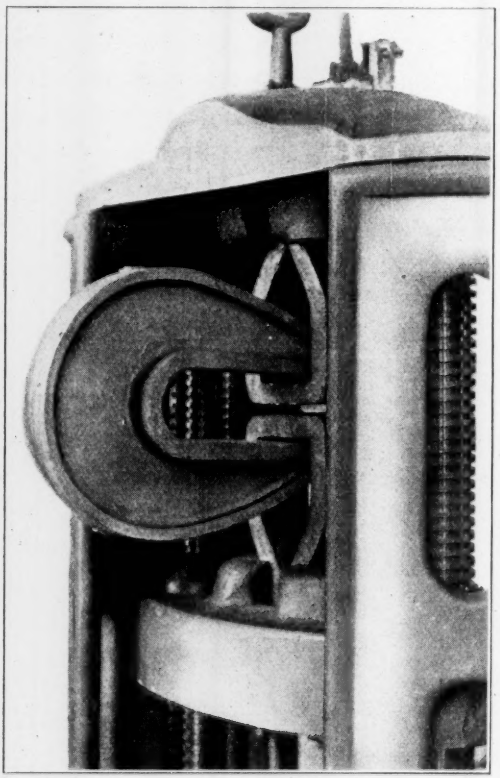

FIG. 6 METHOD EMPLOYED IN APPLYING LOAD

rial. The absurd results given by the Résal and Pearson-Andrews formulas in this case show that they are not applicable to this condition.

14 Castings No. 4, 5, 7, 9 and 12, Plates I and II, were cast from the same pattern and the average dimensions are given in Fig. 2. None of the above-mentioned formulas apply to these castings owing to the peculiar manner in which they failed. The results, however,

have an extremely important bearing on the design of punch frames. These specimens were supposed to fail under a load of about 15,000 lb. When the load on No. 4 reached 9300 lb. a fracture occurred behind the inner flange, as shown in the photographs of castings No. 7 and 12. Continued application of the load produced the fractures shown on No. 4, 5 and 9. The stress producing failure was evidently a normal stress as there is no shear on this section.

TABLE 1 RESULTS OF TESTS

	Tension Specimens			Transverse Specimens Length 12 in.				Curved Specimens			
									Unit stress at *ab*		
Test No.	Breaking Load	Area Sq. In.	Stress Pounds Per Sq. In.	Breaking Load	Area (Breadth × Depth)	$S = \frac{Mc}{I}$	$K = \frac{bd^2}{6M}$	Breaking Load	Barth Formula	Reuleaux's Formula	Pearson-Andrews Formula
1	15000	0.785	19100	2380	.75 × 1.25	36560	3.14	11200	16240	27025	33013
2	14630	0.785	18620	2880	.75 × 1.25	44200	2.52	11125	16120	26844	32796
3	13460	0.709	19000	1680	1.25 × .75	46080	2.65	11390	16540	27484	33577
4	17000	0.785	21630	2280	1 × 1.05	37200	3.48	9300	11330	19583	38223
5	17000	0.785	21630	2000	1 × .95	40000	3.25	8500	10500	17909	34935
6	15500	0.833	18600	2160	1 × 1	39000	2.86	12600	22520	25870	29460
7	16300	0.866	18750	2380	1 × 1	43000	2.62	12000	9790	12584	27048
8	15070	0.724	21700	2310	1 × .95	46250	2.82	15300	12600	18420	
9	18000	0.785	22920	2200	1 × 1	39600	3.07	8300	10130	19164	36530
10	16000	0.785	20370	2300	.95 × 1	43700	2.70	8400	10520		
11	17800	0.754	23600	2060	1.02 × 1	36400	3.94	5200	18420	21040	
12	18000	0.785	23000	2070	1 × .98	38000	3.56	8400	10235	17500	34524
13	18000	0.754	24400	2440	1 × .98	45000	3.20	5800			
14	16300	0.739	21800	2300	1.02 × 1	40600	3.22	12700	23920	26435	28700
15	16100	0.754	21400	2240	1 × 1	40400	3.18	12500	23400	26025	28250
16	16730	0.785	21270	2460	.75 × 1.25	37800	3.41	11255	16320	27160	33300
17	17340	0.785	22080	2741	.75 × 1.25	42200	3.17	11980	17270	28987	35400
18	17900	0.785	22800	2680	.75 × 1.25	41300	3.35	10600	21476		

15 The conditions of stress that would produce such a failure may be given as follows.

a The inner flange being heavier than the web would tend to produce separation behind the flange due to unequal contraction when cooling. This initial stress due to cooling, plus the stress due to lateral contraction caused by the tensile stress due to bending, is a stress normal to the plane of

fracture. This method gives about 1500 lb. for the stress due to the load and about 18,000 lb. for the initial stress due to cooling, which is considered absurd. Hence this method does not offer a satisfactory explanation.

b In Fig. 7, given in the appendix, the load W produces a bending moment at the section AD equal to WN. This moment is resisted by the moment $\frac{SI}{c}$ where S is the sterss at A, I the moment of inertia of the section between A and D, and c the distance between the center of gravity of this section and the point A. The fracture of casting No. 5 suggested this method as a possible cause of failure. At a load of 8500 lb. a fracture occurred behind the inner flange and the load dropped almost to zero. Upon further application of the load the crack gradually approached the outer flange, then the inner flange fractured and finally the outer flange separated. According to this method of analysis the average unit stress at A in the five castings was about 22,700 lb. when the fracture occurred, which is excessive for tensile strength. This does not consider a possible shifting of the neutral axis, however, which would tend to decrease the value of the result. There are also reasons for using a larger value for N. By changing the values of N and c, it is possible to change the value for the stress, hence this method could not be relied upon without making many tests on different-sized specimens with different thicknesses of flange.

c Considering the flanges as separate members having pin connections at points A, E, F and G in Fig. 8, the load W would tend to increase the distance between the points A and G. If A and G be connected by means of a link, the stress in it may be found as follows: Draw the lines EG, FG, EA, and FA and let them represent links with pin connections. Then produce GA until it cuts the line of application of the load at K. The triangle AKE may be taken as the force diagram where $KE = \frac{W}{2}$ and KA is equal to half the stress in the link AG. In the specimen, this stress is taken by the web, the area of which is the outside

diameter, *pp*, of the inner flange multiplied by the thickness of the web. According to this method the unit stress is about 18,000 lb., which seems to express the stress relation very satisfactorily for this particular case, but it is not probable that such results would be derived from castings having a thicker flange.

16 Casting No. 7, Plate II, was first subjected to a compressive load acting on the horizontal side of the notches cut in the outer flange. This caused failure near the line of application of the load, as shown in the photograph. It was then subjected to a tensile load acting at a distance of $5\frac{3}{16}$ in. from the inside of the spine.

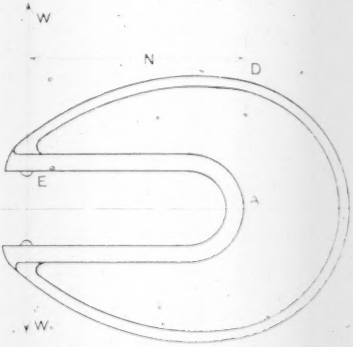

FIG. 7 DIAGRAM SHOWING CONDITION WITH BENDING MOMENT AT SECTION *AD*

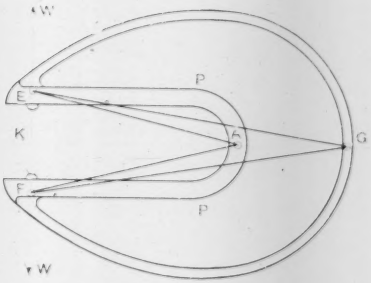

FIG. 8 DIAGRAM SHOWING CONDITION PRODUCING TENSILE STRESS IN SECTION *AG*

17 Casting No. 9 had $\frac{3}{16}$ in. of the outer flange removed on both sides. This, however, did not seem to affect its strength.

18 Casting No. 12 had the outer flange removed on both sides and the inner flange reduced to 1 in. wide at a distance of $6\frac{1}{2}$ in. from the load line; but this did not seem to affect its strength. The unit stress at the milled section according to the formula $S = \frac{Mc}{I}$ was 34,805 lb.

19 Casting No. 10 had the inner flange reduced to $2\frac{1}{2}$ in. wide and failed horizontally as shown. Failure in this manner was probably due to a flaw in the inner flange that extended into the web.

20 Casting No. 11 had about 2 in. of the outer flange removed on both sides and the width of the inner flange reduced to 2 in.

21 Casting No. 13, Plate III, had both flanges removed at a distance of 6.8 in. from the line of application of the load. The stress at

this point according to the formula $S = \frac{Mc}{I}$ is 31,160 lb. per square inch.

22 Casting No. 6 is the same as shown in Fig. 2 with the exception of having the thickness of the web increased to 0.93 in. The fracture occurred at an angle of about 30 deg. with the horizontal. There seemed to be a tendency for the fracture to follow the inner flange.

23 Casting No. 8 was subjected to a compressive load and failed at a section 7 in. from the line of application of the load, the stress being 35,370 lb. according to the formula for beams.

24 Casting No. 14 had the thickness of the web increased to 0.93 in. and provided with large triangular fillets behind the inner flange. By comparing the results of this test with those from No. 6, it seems that enlarging the fillet does not appreciably affect the strength of the casting.

25 Casting No. 15 is the same as No. 14 with the outer flange removed. The results seem to show that the outer flange on No. 14 adds but little to its strength.

26 Casting No. 16 failed in a curved portion. The angle between the plane of fracture and line of application of the load is about 79 deg. The stress in this section by Formula 7 (see Appendix) was 16,300.

27 Casting No. 17 failed in a curved portion, the angle being about 48 deg. The stress in the fractured section by Formula 7 (see Appendix) was 17,300.

28 Casting No. 18 is the same as No. 16 and No. 17 with the depth of spline reduced to 3.4 in. The stress at fracture was 21,476.

THE AUTOGRAPHIC RECORDER

29 The autographic recorder gave an apparently straight line for the deflection-load diagram for each specimen, but the scale of the curve is so small that a slight curvature due to the elastic law would not be perceptible. It is interesting to know, however, that the curve is so nearly a straight line and does not show any change in the elastic law which might be taken as the elastic limit of the specimen.

30 The load-deflection curve shows the total deflection produced by any given load on the specimen, and a sudden change in its direction would indicate that the stress in some section had reached the yield point. Any method of analysis involving the use of these curves is only applicable to the section which yields first. In the case of steel specimens it is difficult to determine this section, and in the analysis of stresses in cast-iron specimens the diagram finds no application.

CONCLUSIONS

31 Although these experiments are not sufficiently exhaustive to render any rigid conclusions, they seem to indicate that the following statements are approximately true:

a There is no rational method for predicting the strength of curved cast iron beams suitable for punch and shear frames.

b Of the three formulas suggested for the design of punch frames, the well known beam formula,

$$S = \frac{Mc}{I} + \frac{W}{R}$$

is the most accurate statement of the law of stress relations existing in such specimens.

c The stress behind the inner flange at the curved portion is an important consideration that should be recognized by the designer.

d There seems to be no definite relation existing between the strength of a curved cast-iron beam and the transverse strength of a test bar cast with it.

e The Résal and Pearson-Andrews formulas are unwieldy and awkward in their application and offer many chances for error.

Acknowledgment is due to Prof. S. E. Slocum for criticisms during the preparation of this article and to Prof. John T. Faig whose assistance made possible the securing of the materials necessary for conducting the tests.

APPENDIX

ELASTIC LAW

Within the elastic limit, homogeneous materials such as steel and wrought iron conform with Hook's law, which states that the stress varies directly as the strain; hence the stress-strain diagram is a straight line. In common practice the elastic limit for cast iron is assumed to be about 6000 lb. per square inch and Hook's law is supposed to be practically true for stresses not exceeding this value but, as a matter of fact, cast iron has no definitely defined elastic limit like steel, and no portion of the stress-strain diagram is a straight line. The elastic law in this case may be expressed by the exponential equation

$$\Delta = K S^m$$

Where Δ denotes the unit deformation, S the unit stress, and K and m are constants. From the experiments of Bach these constants have been determined for cast iron, and the elastic laws found to be as follows:

$$\text{For tension, } \Delta = 0.00001111\ S^{1.0663}$$

$$\text{For compression, } \Delta = 0.00001444\ S^{1.0395}$$

2 These experimental curves may be replaced by parabolas without introducing any serious error. From experiments by Hodgkinson, the equations of the parabolas which fit these stress-strain diagrams most closely were found to take the form

$$S = 1{,}400{,}000\ \Delta\ (1 - 209\ \Delta) \text{ for tension}$$

$$S = 1{,}300{,}000\ \Delta\ (1 - 40\ \Delta) \text{ for compression}$$

3 By assuming that the equation of the elastic law is a parabola, the stress in a beam at a distance y from the neutral axis was expressed by St. Venant in the equations,

$$S = T\left[1 - \left(1 - \frac{y}{c}\right)^m\right] \text{ for tension}$$

$$S' = C\left[1 - \left(1 - \frac{y}{c'}\right)^m\right] \text{ for compression}$$

where T and C denote the ultimate tensile and compressive strengths, c and c' the distance from the neutral axis to the extreme fiber in tension and compression respectively, and m a constant so chosen as to make the parabola fit the stress-strain diagram for the given material. For the value $m = 1$, these equations reduce to the form

$$S = T\frac{y}{c}$$

which is Hook's law.

POSITION OF THE NEUTRAL AXIS AND THE MOMENT OF RESISTANCE

4 The equation

$$M = \frac{SI}{c}$$

is commonly known as the fundamental formula for beams. This formula is due to Navier and is based on the following assumptions: first, the strain is directly proportional to the distance from the neutral axis; second, the neutral axis coincides with the gravity axis; third, the stress is directly proportional to the strain on both the tension and compression sides. Some authorities favor the validity of the first assumption, but they all in general agree that the second and third are not true for cast iron.

5 The fact that the flexural strength given by the above formula does not coincide with the tensile strength has reflected discredit upon it. The flexural strength is from 1.5 to 2 times the tensile strength for ordinary test bars. The cause of this discrepancy is frequently explained by saying that the formula is based on Hook's law, and is not expected to hold true above the elastic limit, hence it does not give the actual tensile strength at rupture. This statement, however, is not necessarily true, because the formula will represent the stress relations in a rectangular beam under all conditions of load provided the coefficients of elasticity for tension and compression are equal.

6 The discrepancy is due to the shifting of the neutral axis which is caused by the difference in the elastic laws for tension and compression. A method for determining the position of the neutral axis is given by St. Venant in his notes on Navier's Resistance of Solid Bodies. Assuming that the strain varies directly as the distance from the neutral axis, the stress will vary as in the ordinary stress-strain diagram. Now if the beam fails in tension, the stress at rupture on the extreme fiber on the tension side is equal to the ultimate strength of the material.

7 Let $BE = S_t$ represent the ultimate tensile strength of the material, and let OE and ODF represent the stress-strain diagrams for tension and compression respectively as shown in Fig. 1. Then OB is the distance of the neutral axis from the tension side of the beam drawn to the same scale as yet undetermined, and the area OEB represents the total tensile stress in the beam at this section. But since the total tensile and compressive stresses must be equal, the compressive side of the beam is found by drawing a line AD such that the area OAD is equal to the area OEB. OA will then represent the distance of the compressive side of the beam from the neutral axis to the same scale as that to which OB is drawn. The scale may be easily determined by comparing AB with the actual depth of the beam.

8 The stress-strain diagram for the compression side is more nearly a straight line than that of the tension side. By assuming that the compression side obeys Hook's law, the formula for the moment of resistance is thereby simplified and becomes for a rectangular section of breadth b and depth d,

$$M = S\frac{bd^2}{6}\left| \frac{m\left(\frac{3(m+3)}{m+2} + 1 - \sqrt{\frac{2}{m+1}}\right)}{m+3+2\sqrt{2(m+1)}} \right|$$

The value of m in this equation is to be so chosen as to make the expression for the elastic law conform as closely as possible with the actual stress-strain diagram. This equation for any given material reduces to the form

$$S = \frac{MK}{bd^2}$$

9 Hodgkinson's[1] experiments led him to conclude that the neutral axis did not coincide with the gravity axis just at the point of failure of a cast-iron beam, that it moved toward the compression side, dividing the depth into a ratio of 1:5 or 1:6, which is the ratio of the ultimate strengths of the material in tension and compression.

10 W. H. Barlow[2] experimented with beams 7 ft. long, 6 in. deep and 2 in. wide, and found that loads less than three-fourths of the breaking load did not change the neutral axis materially, but just before rupture the shifting became greatly increased. These experiments accounted for only a small percentage of the discrepancy assumed by Hodgkinson.

11 Another explanation is that the outer fibers are subjected to an initial compressive stress due to their being cooled before the inner portion of the beam, and this initial stress must be overcome before the tensile stress begins.

12 Lewiston[3] proposed the theory that the compression side of a cast-iron beam is stressed to its ultimate strength at the point of failure. That is to say, a beam is stressed to its ultimate strength on both the compressive and tensile sides just before failure occurs. According to his theory, the distance from the neutral axis to the fiber that fails first is

$$N = \frac{I}{C + T}\, d$$

where C and T are the ultimate strengths in compression and tension and d the depth of the beam. As a result of this theory the moment of resistance of a rectangular cast-iron beam is

$$M = \frac{5}{18}\, S\, bd^2$$

when the ratio of the ultimate strengths, $\frac{C}{T}$, is equal to 5. This equation may also be written

$$S = \frac{MK}{bd^2}$$

where S denotes the ultimate strength of the material in tension.

13 Emery[4] claims that neither Hook's law nor Bernoulli's assumption are true for cast iron, that the medial portion between the neutral axis and the extreme fibers is stressed more than assumed by Bernoulli's assumption, which would necessarily relieve the outer fiber of a certain amount of stress. In devel-

[1] Hodgkinson: Experimental Researches on Cast Iron.
[2] Phil. Mag., 1855.
[3] Trans. Am. Soc. C. E., Vol. 35.
[4] Trans. A. S. M. E. Vol. 8.

oping a formula for the resistance of a rectangular cast-iron beam he assumes the elastic law,

$$J = KS^m$$

for the tension side and Hook's law for the compressive side, which gives

$$M = S\,bd^2 \frac{\left(n + \frac{2}{3}\right)\left(2\,m\,t\right)}{\left(1 + 1\,2\,m\,t\right)^2}$$

where S is the tensile strength of the material, n and m are constants from integration and t the ratio into which the neutral axis divides the depth of the beam. This equation may also be expressed

$$S = \frac{MK}{bd^2}$$

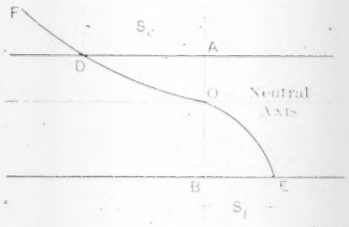

FIG. 1 STRESS-STRAIN DIAGRAM FOR TENSION AND COMPRESSION

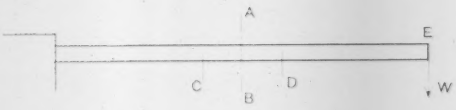

FIG. 2 CANTILEVER BEAM SUBJECTED TO A BENDING MOMENT

The values of K given by Mr. Emery range from 2.8 to 4.

14 Clark's formula,

$$S = \frac{WL}{1.55\,bd^2}$$

gives a value of $K = 3.46$.

15 Keep[a] published the results of a great many tests of different-sized specimens made of different grades of cast-iron and concludes that the silicon content and the rate of cooling greatly affects the strength of cast-iron. He also points out the fact that there is definite relation existing between the shrinkage and the percentage of silicon, and by measuring the shrinkage of a bar, the silicon content may be found from his shrinkage chart. This same chart is also provided with curves from which the strength of any bar between $\frac{1}{2}$ in. and 4 in square may be found when the strength of any other bar within these limits is known. In the discussion of Mr. Keep's paper, Professor Benjamin proposed the formula

$$W = H\frac{b^{0.83}\,d^{1.89}}{L^{1.058}}$$

[a] Trans. A. S. M. E. Vol. 17.

where W is the breaking load and H a constant determined by breaking a bar 1 in. square and 12 in. long.

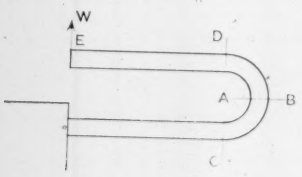

FIG. 3 BENT CANTILEVER BEAM SUBJECTED TO A BENDING MOMENT

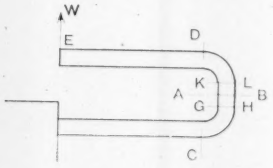

FIG. 4 BENT CANTILEVER BEAM SUBJECTED TO A BENDING MOMENT

16 Although based upon different theories and experiments, most of the above formulæ for straight cast-iron beams of rectangular cross-section reduce to the form

$$S = \frac{MK}{bd^2}$$

For rectangular sections the value of K has been found to vary between 2.8 and 4, depending upon the elastic laws of the material when subjected to tensile and

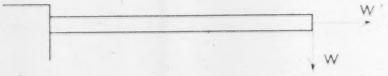

FIG. 5 STRAIGHT BEAM SUBJECTED TO BENDING LOAD

compressive stresses; and these laws vary with the sectional area, chemical composition, temperature of the metal when poured and the rate of cooling.

EFFECT OF CURVATURE AND FORMULAS FOR CURVED BEAMS

17 In Fig. 2 the simple cantilever beam is subjected to a bending moment due to the load W supported at the end. The bending moment at the section AB is resisted by the tensile and compressive stresses in the upper and lower fibers of the beam.

18 Consider the beam when bent into the forms shown in Fig. 3 and Fig. 4. The condition of stress in the portion ED is not affected by the change in shape; whereas the section AB resists not only the bending moment due to the load, but an additional force due to the direct pull of W.

19 The stresses in any section of a straight beam subjected to a bending load W, and a tensile load W' as shown in Fig. 5, may be accurately represented by the well known formulæ,

$$S_t = \frac{Mc}{I} + \frac{W'}{A}$$

$$S_c = \frac{Mc}{I} - \frac{W'}{A}$$

20 These formulæ are supposed to be true for straight portions of beams only; but due to their simple form, they are often applied to curved sections such as AB in Fig. 3; in which case $W' = W$ and $M = WL$, where L is the distance from line of application of the load to the gravity axis. When used in this way these formulas assume that the neutral axis coincides with the gravity axis of the section and its position is not affected by the curvature of the beam. These formulas may also be written in the form,

$$S_t = \frac{W}{A}\left(\frac{Lc}{R^2} + 1\right) \qquad [1]$$

$$S_c = \frac{W}{A}\left(\frac{Lc}{R^2} - 1\right)$$

where R is the radius of gyration and equal to

$$\sqrt{\frac{I}{A}}$$

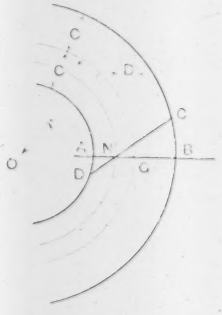

FIG 6 ENLARGEMENT OF CURVED PORTION FIG. 3

21 Fig. 6 shows an enlargement of the curved portion of the beam represented in Fig. 3. Assuming that a plane passed through the section AB before the load is applied remains a plane under the conditions of stress, that the stress is directly proportional to the distance from the neutral axis, and the radius of curvature of the gravity axis does not change; it will take less stress to deform the fibers on the convex side the same amount as on the concave side, because the convex side is longer. When the stresses at A and B are equal, the deformations may be represented by BC and AD respectively. The neutral axis does not coincide with the gravity axis at G, but passes through the point N which is nearer the convex or tension side, and at a distance D from G. The distance D, through which the neutral axis shifts, is independent of the stress when expressed by the equation.[6]

$$D = -\frac{\int \frac{y\,dA}{y + r}}{\int \frac{dA}{y + r}} \qquad [2]$$

[6] This method of analysis is due to Résal and is given in Strength of Materials by Slocum and Hancock.

where y denotes the distance from the gravity axis to any fiber having an area equal to dA, and r the radius of curvature OG of the gravity axis.

22 The stress on any fiber at a distance y from the gravity axis is expressed by the equation

$$S = \frac{M}{(y+r)} \frac{(y+D)}{\int \frac{(y+D)^2}{y+r} dA} \qquad [3]$$

where the bending moment, $M = W(L-D)$.

23 The application of the above formula may be greatly simplified by a geometrical transformation of the section as shown in Fig. 7. Let $KLMN$ represent a section cut by the plane AB in Fig. 3, OS the axis of symmetry passing through the center of curvature O and GP the gravity axis of the section perpendicular to OS. By drawing radial lines from O through each point in the boundary, such as T, cutting the gravity axis at some point R, the lines drawn through T and R, perpendicular to OS and GP respectively will intersect in the point E. The locus of the point E is the boundary of the transformed curve $klmn$.

24 It has been proven by Résal that the distance between the center of gravity G of the original section and the center of gravity G' of the transformed section is equal to the value of D given in Equation 2, and the moment of inertia of the transformed section is equal to the integral in Equation 3. By denoting the moment of inertia of the transformed section by I', the above equation reduces to

$$S = \frac{Mr}{I'} \frac{y+D}{y+r} \qquad [4]$$

25 Another method of analysis is that due to Pearson and Andrews which takes into account the change in the radius of curvature and decrease in cross-sectional area due to lateral deformation. The tensile stress on the concave side is expressed by the formula

$$S = \frac{W}{A}\left[\frac{L}{r\gamma_2}\left(\left(1-\frac{c}{r}\right)^{n+1} - \gamma_1\right) + 1\right] \qquad [5]$$

where

W = load in pounds;
A = area of cross-section in square inches;
L = distance from line of application of load to the gravity axis of the section;
r = radius of curvature of gravity axis in inches;
c = distance from gravity axis to extreme fiber on tension side;
n = Poisson's ratio of lateral contraction.

The value of γ_1 is determined graphically as follows: Let $KLMN$ in Fig. 8 represent the cross-sectional area cut by the plane AB in Fig. 3; r the radius of curvature, GP the gravity axis and y the distance from any point C on the boundary to the gravity axis. For each value of y, or position of D lay off

$$CE = \frac{CD}{\left(1+\frac{y}{r}\right)^{n+1}}$$

forming the boundary *klmn*. The area of *klmn* divided by the area of the section *KLMN* is equal to the value of γ_1.

26 The value of γ_2 is equal to

$$\frac{\text{area of } klmn\text{—area } k'l'm'n'}{\text{area of } KLMN}$$

where the boundary of the area $k'l'm'n'$ is formed by making

$$CJ = \frac{CD}{\left(1 + \frac{y}{r}\right)^n}$$

for each value of y.

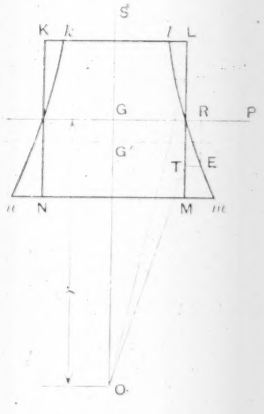

FIG. 7 DEFORMATION OF SECTION *AB* OF FIG. 3 WHEN UNDER STRESS

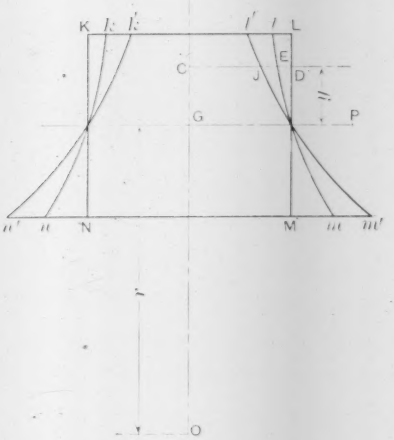

FIG. 8 DEFORMATION OF SECTION, UNDER PEARSON AND ANDREWS THEORY

27 The stress in any section of the straight portion of the beam shown in Fig. 3 may be found from the formula

$$S = \frac{Mc}{I}$$

and the stress in a section perpendicular to the line of application of the load, such as *AB*, may be found from Formula 4, provided the beam is of homogeneous material and obeys Hook's law, but none of the above formulas can be accurately applied to the section between *A* and *C*.

28 It is not necessary, however, to investigate more than one section of the curved portion provided the throat depth is small, as in the case of hooks; but in designing large punch frames it is desirable to determine at least two sections within the curved portion.

29 By comparing the properties of cast iron with the assumptions upon which the Formulas 1, 3, and 4 are based, it is easily seen that they do not apply to cast iron with any degree of accuracy. Hodgkinson's experiments

seem to show that due to the properties of the material, the neutral axis of a cast-iron beam shifts toward the compression side a distance

$$D' = \frac{d}{3}$$

Résal's formula as given in Equation 2 when applied to a rectangular section shows that

$$D = r - \frac{d}{\log_e \frac{2r + d}{2r - d}}$$

30 If the radius of curvature be such as to make $D' = D$, it seems that Formula 1 would apply. By equating the values of D and D' and solving, $r = 0.0505\ d$, which is a value that could not be used in practice. Hence, there is apparently no rational formula that represents the relation between the breaking load and the unit strength of curved portions of cast-iron beams similar in form to that shown in Fig. 3.

31 Press and punch frames requiring a wide gap are usually shaped similarly to the beam shown in Fig. 4, in which case a portion of the spine is straight. The sections KL and GH may be considered as belonging to either the straight or curved portions. If considered as belonging to the curved portion formula 4 should apply; but if taken as belonging to the straight portion it is subjected to the same conditions of stress as the section AB, and formula 1 should be applicable. These formulas, however, give different results which seem to indicate that the section AB should be less than the sections KL and GH of the curved portions, hence the condition of stress at KL and GH is somewhat similar to that due to sudden change in cross section, or which exists in the section connecting a spherical end to a thick cylinder.

32 A formula for determining the ultimate strength of castings having straight spines should recognize the change of the neutral axis due to the elastic properties of cast-iron. Such a formula based on the knowledge of straight beams of rectangular section takes the form

$$S = \frac{MK}{bd^2} + \frac{W}{A} \qquad [6]$$

where K is a constant to be determined experimentally.

33 The stress at sections between A and D in Fig. 3 and in sections above KL in Fig. 4 are usually determined by the formula

$$S = \frac{Mc}{1} + \frac{W \sin \theta}{A} \qquad [7]$$

where M is the product of the load into the distance between the line of application of the load and the gravity axis of the section considered; and θ the angle between the line of application of the load and the section produced until they intersect. For rectangular sections this formula may also be written

$$S = \frac{MK}{bd^2} + \frac{W \sin \theta}{A} \qquad [8]$$

FIRES: EFFECTS ON BUILDING MATERIAL AND PERMANENT ELIMINATION

By Frank B. Gilbreth, New York

Member of the Society

A remarkable paper, Bulletin 418, just issued by the United States Geological Survey, entitled The fire tax and waste of structural materialism in the United States, by Herbert M. Wilson and John L. Cochrane, contains the following astounding data:

The total cost of fires in the United States in 1907 amounted to almost one-half the cost of new buildings constructed in the country for the year. The total cost of the fires, excluding that of forest fires and marine losses, but including excess cost of fire protection due to bad construction, and excess premiums over insurance paid, amounted to over $456,485,000, a tax on the people exceeding the total value of the gold, silver, copper, and petroleum, produced in the United States in that year.

The actual fire losses due to the destruction of buildings and their contents amounted to $215,084,709, a per capita loss for the United States of $2.51. The per capita losses in the cities of the six leading European countries amounted to but 33 cents, or about one-eighth of the per capita loss sustained in the United States. In addition to this waste of wealth and natural resources, 1449 persons were killed and 5654 were injured in fires.

2 In discussing this waste, Charles Whiting Baker, editor of Engineering News, in an address before the meeting of the national engineering societies on the Conservation of Natural Resources, March 24, 1909, said:

The buildings consumed, if placed on lots of 65 ft. frontage, would line both sides of a street extending from New York to Chicago. A person journeying along this street of desolation would pass in every thousand feet a ruin from which an injured person was taken. At every three-quarters of a mile in this journey he would encounter the charred remains of a human being who had been burned to death.

The American Society of Mechanical Engineers, 29 W. 39th Street, New York. All papers are subject to revision.

The results obtained indicate that the total annual cost of fires in the United States, if buildings were as nearly fireproof as in Europe, would be $90,000,000, and therefore that the United States is paying annually a preventable tax of more than $366,000,000, or nearly enough to build a Panama canal each year.

It will be incidentally noted that fire protection involves the use of 2,000,000 tons of metal, having a value in excess of $127,000,000, and the metal in 350,000 hydrants, having a value of $30,000,000, all of which is wasted on account of the need of preparing to fight fires of a kind which, because of the inflammable character of building construction in this country, would develop into conflagrations without adequate water service and fire departments.

The estimated cost of private fire protection, including capital invested in construction and equipment, aggregates about $50,000,000, and the annual interest on this sum and the annual cost of watchmen's services amount to about $18,000,000.

3 In this age of "conservation" it is amazing that so little has been done to prevent destruction by fire, and to apply the lessons that every great fire teaches us. The interest of the entire public in fireproof construction flares up with a great fire, but flickers out before the building permits for new structures are granted. The average owner when rebuilding even finds fault with the authorities whose duty it is to enforce the inadequate existing ordinances relating to noncombustible construction and to "fire-stopping" in combustible buildings.

4 There must probably be more big fires among buildings now built, but there is no sane excuse for any such loss in the buildings to be built in the future. Within the last six years there have been wonderful opportunities for studying great fires here in our own country. It has been possible to observe the destruction that occurred, to discover why it occurred, and to determine exactly what must be done if repetition of such destruction is to be prevented in the future.

5 The writer has been fortunate in being able to make careful observation of the great fires at Toronto, Sioux City, Baltimore, San Francisco, Chelsea and elsewhere. Having become interested in the effects of fires as observed in Sioux City and Baltimore, the observations in San Francisco were made with great care and detail. The ruins of each of the 503 city blocks in the San Francisco fire were visited and studied. This led to the formulation of definite conclusions which have since been again checked by the Chelsea fire. The conclusions here presented, therefore, are not the result of theorizing or reading only, but of painstaking and directed observation.

6 All great fires are alike. Building material behaves the same

in a fire regardless of the location. We will therefore study a typical building in a typical fire.

7 The building of the Mutual Life Insurance Company at San Francisco was a steel frame structure, eight stories high, of the best construction in 1892, when it was erected. The laying up and filling of the joints in the brick, stone and terra cotta were as nearly perfect as possible. The exterior wall completely enclosed the steel frame, which was put together with bolted connections. The floors were of hollow terra cotta flat arches, and the partitions were hollow terra cotta blocks. The damage to the building, which necessitated the removal of the upper six stories, was practically all done by fire.

8 This building is excellent for an illustration, because it shows the good and bad points of many different kinds of incombustible materials, which were used in its construction. We took it down after the fire, photographing every part as it was taken down.

9 Examinations while taking down a building give the ideal method of studying the cause and effect and possible remedy for damage by fire.

10 The pictures illustrate better than could any description the process and the stages of removal. They show also that a very small quantity of wood in a so-called "fireproof" building built almost entirely of non-combustible materials will furnish sufficient heat to destroy it. The lessons from this and from all fires point to these conclusions: *No structure of the future should be built of wood. No structure of the future should contain any wood.* It has taken costly lessons to teach us this, but if we have at last learned the lesson, the price we have paid is cheap.

11 It has long been realized that, with the wood supply of the world constantly diminishing, wood was bound to become too expensive to use as a building material. But it is not so widely realized that the day when wood is as costly as non-combustible building materials is *here*. Today, concrete structures, and in some localities other non-combustible structures, can be erected at no greater first cost than wooden ones; with an added element of safety to those who use the building which cannot easily be overestimated.

Cost of Concrete Construction

12 In order to hold the place it claims—that of the cheapest and best fireproof construction material—concrete must be able to show itself cheapest in first cost, as well as cheapest in the long run.

13 Concrete materials are obtainable everywhere: stone, ledges, gravel, sand, burnt ballast, slag, brick refuse, terra cotta chips, stone cutters' refuse and cinders, are all good aggregates for concrete.

14 As far as the cost of the labor of putting the material in place is concerned, much has been done toward reducing that to a minimum. With metal forms and certified concrete, concrete construction is cheaper today than that with brick, hollow terra cotta tile or any other material.

15 Charles T. Main, a member of the Society, has recently published the most valuable contribution regarding cost data of brick and wood mill construction. Mr. Main is a leading authority on this subject, and his data are conceded to be correct. Mills have been built entirely of concrete, in all parts of the country, in less time than it is possible to build them of brick and wood and for less money than the square foot and cubic foot prices collected by Mr. Main. A few years ago this statement would not have been true. Today, with the aid of new designs, new methods, new forms, scientific grading and proportioning, it is a fact that cannot be disputed, but one, nevertheless, not generally known.

16 The development of the entire concrete building and of the parts of the concrete building have been slow, because it has taken a long time to learn to *think* in concrete. Each student brought his stock of experiences from another field to bear on the problem, but the problem was so new that few could understand it in all its aspects. The engineer, who knew how to design the strength, knew little about artistic features. The architect, who alone could give beauty to the building, was not an expert in the methods that make low costs. Progress was made more difficult by the fact that so few of the structural features and construction methods were standardized.

17 The contractor, having analyzed his costs, saw that the forms were the largest single item in the cost of the concrete. This fact had its effect on the design of the concrete building, in the trend toward straightway repetition form work and flat-slab ceilings, with as little beam and girder work as possible. A further benefit was in the fewer corners for fire to attack or for interruption of the action of sprinklers, saving also in the cost of the sprinkler system.

18 After the flat-slab ribless type of ceiling, with its low cost of forms, came standardization of the units of forms for walls and columns. With the use of these metal form-units a new problem faced us, namely, slight modification of design to agree with the cheapest labor requirements of the metal forms. The influence was

also felt, of the fire protection design thoroughly worked out and standardized by Charles L. Norton, professor of heat measurements at the Massachusetts Institute of Technology and engineer in charge of the engineering experiment station of the Factory Mutual Fire Insurance Companies. Mr. Norton's recommended practice, which is now universally accepted, is so simple and apparently so obvious that we sometimes forget to give him the credit for his work.

19 The importance of the forms, from the standpoint of economy, in being able to build fireproof buildings for the price of brick and wooden mill construction, is clearly shown by the fact that Charles D. Watson, of Syracuse, N. Y., on account of the reduction in the number and expense of forms, has been able to build a number of buildings from separately cast members and reduce the costs below anything ever cast *in situ* in wooden forms.

USE OF METAL FORMS

20 We come now to the latest developments, the metal form-units that have been perfected, densifiers for compacting the concrete, mechanical conveyors for displacing the wheel-barrow and hand tamping, and the concrete-producing factory, centrally located. These things have all arrived. Metal molds have reduced the price of concrete to that of wooden construction.

21 For the greatest saving in the cost per cubic foot of the completed building, the methods of construction, as well as the designs, must be standardized. This is rapidly being done. For example, the metal forms are set up and concrete is hauled in large quantities from a centrally located factory and poured at once for the entire story of a building. This does away with the joints between the work of two different days, and permits of figuring the tensile strength of the concrete in design calculation, with a great consequent saving of reinforcement. Such calculations cannot be safely figured unless the whole story is poured at once.

BEARING ON ULTIMATE COST OF HEALTH CONSIDERATIONS

22 Figuring the cost of furring, lathing, plastering, interest, fixed charges, heating, and repairs, concrete is certainly cheaper than wood. If you also figure health, concrete is *very much* cheaper. A concrete building is cooler in summer and warmer in winter than any other kind. It can be cleaned out with a hose. It can be easily, quickly and thoroughly fumigated, room by room, or all at once. It is drier

than any other, provided it is built with self-ventilating air spaces in all of the exterior walls. A concrete building is vermin-proof, and should germs or vermin get in the building, it may be easily rid of them.

23 The 1907 Year Book of the Department of Agriculture says (Page 98):

> No one of our wild mammals, possibly not all combined, does as much damage as the common rat. . . . .
>
> The rat continues to cause great losses throughout the United States. During the past year an attempt was made to ascertain the approximate damage done to property by this rodent, in the cities of Washington and Baltimore. Many business men were interviewed, including dealers in various kinds of merchandise, feeders of horses, managers of hotels and restaurants, and manufacturers. The inquiries included all sections of the two cities, and both small and large dealers. Data were obtained from some six hundred firms and individuals, from which it was estimated that the annual loss from rats in Washington is about $400,000; in Baltimore, upwards of $700,000. Assuming, as is probable, that similar conditions obtain in all our cities of over 100,000 inhabitants, the damage by rats in these centers of population entails a direct loss of $20,000,000 annually. This enormous sum gives an idea of the still greater total loss inflicted by this rodent throughout the length and breadth of the land.
>
> The rat continues also to excite grave apprehension, because of its agency in distributing the dreaded plague and other diseases. Boards of Health and the Marine Hospital Service in several of our maritime cities have been prosecuting active war against the rodents, and large sums have been expended in efforts to effect their extirpation. No one method has proved adequate, and only by concerted, systematic, and persistent efforts is it possible to reduce and keep down their numbers. The rat-proof construction of buildings, the constant use of traps, and the use of poisons wherever possible, will go far toward assuring public safety. Experiments with various poisons and mechanical means of destruction have been made during the year, and a report of the subject with recommendations will soon be issued.

24 If wooden buildings are to be tolerated because of their alleged low cost, consider then that the concrete structure furnishes no hiding places for the rats and mice that cause damage amounting to a large rate of interest on over a billion dollars annually. There is no danger in exterminating rats by wholesale poisoning in concrete buildings, for there are no cracks for them to crawl into and die.

Advantage of Concrete Construction in Elimination of Fire Loss

25 It may seem that we have wandered from our subject of fires and their permanent elimination, but it was necessary to show that concrete is thus, from *every* standpoint, cheaper than wood, its

only rival for low cost. Concrete construction is the best form for the elimination of fires, because it affords:

a Least cause for fires.
b Least amount of damage to structural parts by fire.
c Least amount of damage to structural parts by water.
d Least amount of damage to contents of building by water.
e Least quantity of combustible structural materials in a room.
f Least speed of combustion of the contents of a room
g No concealed fires; all fires are in plain sight.
h Least spread of fire to adjoining parts of the same story.
i Least spread of fire to stories above.
j Least spread of fire to next buildings.

26 The causes of fires this paper will not discuss. The amount of damage done by a fire in a concrete building depends upon circumstances perfectly within our control and predeterminable. With concrete made of properly selected fire-resisting materials, practically no damage is done, except by prolonged high temperature. The results of recent tests by Prof. Ira H. Woolson, a member of the the Society, and his assistant, J. S. Macgregor, prove conclusively that a concrete building properly designed, with as few projecting corners as possible, will withstand long periods of the hottest hardwood fires, with no resulting damage that cannot be thoroughly repaired with mortar and a plasterer's trowel. These tests were on full-sized rooms with walls of concrete made of different kinds of material.

27 Concrete for walls can be poured in metal molds with sufficient accuracy to permit of painting or wall-papering without further plastering or smoothing. This means that the best of this fire-resisting material is brought to the very surface of the wall where the flames strike. With metal molds all corners can be made rounded as cheaply as square, or metal corner beads may, if desired, be inserted and retained in the molds. Now, if a fire does occur in a building made of concrete cast in smooth metal forms, the damage is less than in any other type of building and the danger of spreading is less.

28 A concrete building can be damaged by water less than any other form of construction. To begin with, water does not injure concrete; in fact it improves its quality. There is no wood to swell, and afterwards to shrink and crack the plastering. Even the paint best suited to be used on concrete lends itself particularly well to

washing down with water. There are no hollow spaces that the water can flow through, damaging the contents below. A concrete building is water-tight from floor to ceiling, and small quantities of water can be easily handled through small scuppers, either into the air space of the vaulted wall, or through the wall to the outside. The fire is never hidden by the construction; consequently no unnecessary streams of water are flooded into the building.

29 In a concrete structure there need be little or no combustible material. Let us take for example the dwelling house, for it is the most difficult to make both cheap and fire-proof. In a concrete residence there is little trim that cannot be made better and cheaper of portland cement than of wood. The chair rails and picture molding can be made of concrete. The trim around the windows and doors can be molded in metal molds as cheaply as straight members. Even the wire moldings can be done away with, and the conduits buried in the concrete partitions, walls, ceilings and floors. Baseboards should be made of concrete or else omitted entirely—as they serve no useful purpose in a concrete building, except in following wooden precedents. Windows may have cement sashes, with wired glass, and self-closing shutters, or self-dropping shutters of rolled-up metal or asbestos. Metal furniture may be used. The paint and varnish used on buildings and furniture should be selected carefully, as these are great factors in determining the temperature at which a fire will start and the speed with which it will spread. There is also a great difference in the paints and varnish used for painting concrete.

30 The flooring need not be of wood. There are many first-class non-combustible materials beside portland cement that will fill every good requirement of wood and still be fireproof. There are also parquetry floorings, made of slow-burning wood, and much thinner than the old-fashioned hardwood floor, which make the best flooring in case wooden floors are desired. In a concrete residence there is no excuse for wooden under-flooring nor for wooden screeds.

31 The best form of construction for the elimination of fires is one that keeps the fire in plain sight, once it is started. One of the greatest obstacles in fighting fire in wood construction is the difficulty of locating a fire after it is known to exist. Every second of time lost gives it accelerated fury. Holes cut through the walls, floors and ceilings to locate a fire oftentimes only furnish more drafts to feed the flames.

32 With hollow concrete blocks, or monolithic cast-in-place walls

and partitions there are no chances for the fire to be concealed. There is no cause for cutting into the partitions floors or ceilings. There is no damage to the structure from cutting. no injury from water. With such a building, if a fire does start, it starts in plain sight remains in plain sight at all times and can be attacked instantly before it has spread; and the water damage will be practically nothing. Fires in plain sight are usually extinguished before the arrival of the fire department.

Checking the Spread of Fires

33 The spread of a fire into adjoining parts of the same story is possible in a concrete building only through doorways, pipe holes, etc. A concrete wall is an ideal barrier to the spread of any fire. Not only is it incombustible, but it is unaffected for a long period by any but extremely high temperatures. Its coefficient of expansion, 0.0000055 to 0.0000060, is practically the same as that of steel; consequently, there is no damage from unequal expansion of the concrete and its steel reinforcement. As for doorways, any of the makes of fireproof doors and windows well made in accordance with the requirements of the Boston Mutual Fire Insurance Company will stop completely almost any fire. There are metal-covered doors made that very few people can distinguish without touching from mahogany or other hard woods. These doors will confine any ordinary fire to one room in a concrete building; or they will hold any fire in one room long enough to enable the fire to be handled after it has got by the door.

34 The spread of fires in the same story is very slow compared with the spread vertically. Fire may be communicated from one story to another by means of well-ways stairways, or holes in the floors, or by the fire passing out of a window on one story and in at the window above. In a concrete building, the floor forms an ideal fire stop even if the walls have been furred or lathed with wood.

35 There are so many good and cheap metal lathings and furrings on the market today that wood should be prohibited everywhere. While metal does cost more, in the long run it is cheaper to use. With vaulted concrete walls there is no real need of furring or lathing. Concrete construction is particularly well adapted to plastering directly upon the concrete. Furthermore, only one coat is needed to make the same quality of work as two or three coats upon wood or steel lathing. When the plastering is put directly upon the wall, there is no space in which the fire can travel.

36 The regular openings can be protected in the accepted way, and kept free from combustible contents.

37 The spread of a fire to the next building is caused by the combustible gases distilled by the heat from the paints, and varnishes, and even from the wood itself. This explosion (which firemen call hot-air explosion) sets the building on fire in several places at once. While the contents of a concrete building may burn, such a fire will not ordinarily make heat enough to cause the fire to jump wide spaces between buildings. As a further guard the best window, whether translucent or transparent, is one of wired glass. The wires should extend out from the glass sufficiently to be embedded in the concrete sash, holding the glass in place, even when it is heated to a point where a sash of other material would drop the glass or where the glass would fold down and lose its former shape.

38 The disuse of wood in building construction will mean:

a Saving of forests.
b Uninterrupted business.
c Saving of life.
d Saving of buildings.
e Saving of the contents of buildings.

Government Aid in Advancing Fireproof Construction

39 The work to be done to reduce fire losses to a minimum is so great and so important that it can never be thoroughly and completely done until the Government takes it in hand. The Government could aid fireproof construction in various ways.

a By passing laws restricting the use of wood in buildings.
b By levying taxes, discriminating in favor of fireproof houses and against wood in construction.
c By educating the people by Government documents on how to build fireproof houses.
d By establishing a Government bureau for disseminating information regarding honest unbiased fire tests on material, together with Government experiments on different full-size buildings,—kinds, types, materials, etc.—with bulletins of the progress. The cost of this would be but a trifle compared with the benefits. There are several men specially qualified to advise on this work,

as they have been doing such work in their own laboratories for years.

(e) By building fireproof houses for the use of Government Departments and disseminating information concerning them by means of bulletins.

40 In the meantime, prize competitions would cause special investigation and study in all parts of America for the best house. A few attempts at such competitions have been made in the past. The judges have generally awarded the prizes, in accordance with their best ability, to the designer whose plan pleased them the most. Instead of this, the award should have been made to the competitor who could show the lowest cost per cubic foot, for the finished house with certain specified fireproof requirements. After the standard requirements of a house have been fulfilled, and developed in accordance with the best scheme of fire protection, durability, low cost, permanency speed in building, and comforts in winter and in summer, it is then time to add those beautifying effects that are so necessary from an art standpoint. Such a competition would show to the amazement of many that a better house can we build more cheaply of incombustible materials than of wood.

CONCLUSIONS

41 The only excuse for using wood construction is its low cost. Today—here—now—we have an incombustible material, a material at a less cost than wood that has stood and will stand high temperatures for long periods without injury. Wood must not be used. We do not argue that no other non-combustible material shall be used, and that concrete shall be used exclusively. There are many cases where other non-combustible materials have special merits. They should be used when it is advisable to do so. But now that we have a cheaper and incombustible substitute for wood, wood construction, wood trim and wood finish should be legislated and taxed until wood is eliminated from all building construction.

FIG. 1 WEST ELEVATION

FIG. 2 EAST ELEVATION

Note that the granite was ruined where flames came out at openings. Where the draft was inward, the granite was in perfect condition. The ornamental terra cotta, though apparently but little injured, was completely ruined by heat cracks. Note crack entire height of building, visible in cornice (Fig. 1), showing line where bolts were completely sheared at each story by expansion due to heat.

Cracks extending from top to bottom, about on a line with the clay flue (Fig. 2) mark the location where the bolts sheared. Note also the cracks in the east wall at the fifth-floor level.

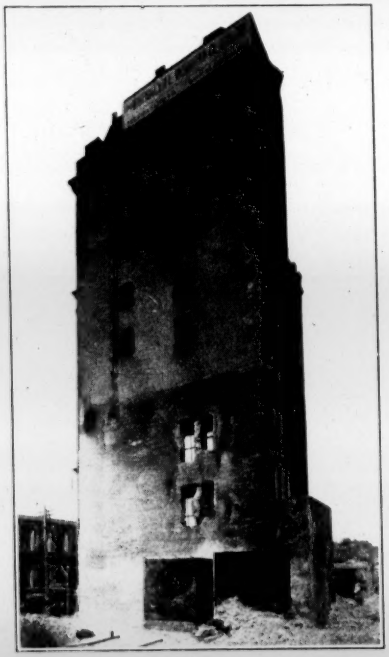

FIG. 3 SOUTH ELEVATION

FIG. 4 FALLEN ROOF TRUSS

Showing complete destruction of adjoining buildings. Structural columns in top story pulled out and broken.

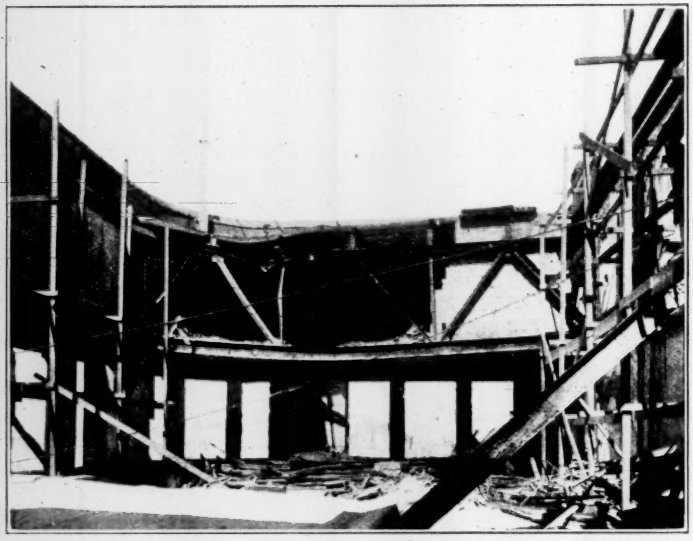

FIG. 5 VIEW LOOKING NORTH IN TOP STORY

Note the complete destruction of the roof trusses and various kinds of material, and that the wooden flagpole was not even scorched.

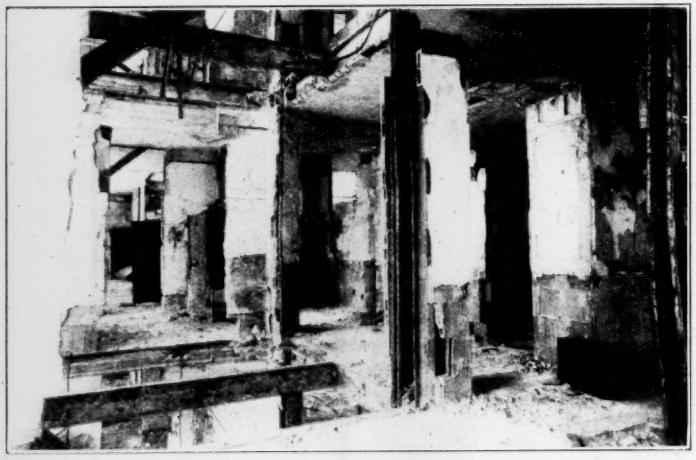

FIG. 6 DAMAGE TO ELEVATOR FRAMING

Although the only combustible material in this building was the wooden trim, floor boards, screeds and furniture, there was sufficient heat to warp the cast iron around the elevators out of shape and to burn up the contents of the steel vaults which were in nearly every room.

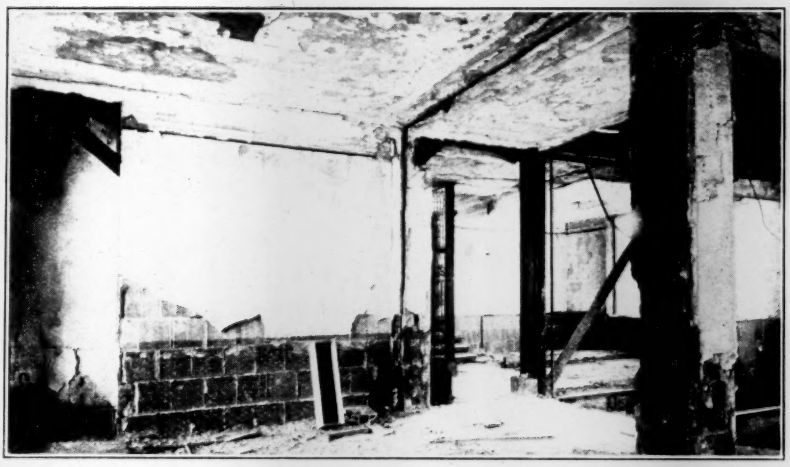

FIG. 7 DESTRUCTION OF NON-COMBUSTIBLE MATERIAL

In all these pictures the columns having pipes buried behind the fireproofing have variably had the fireproofing pushed away from the columns, due to the expansion of the pipes by heat. Note the complete destruction of the marble dado.

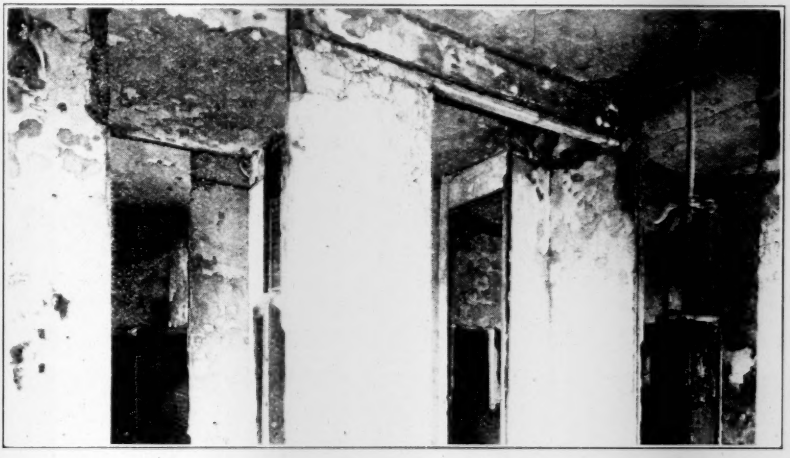

FIG. 8 COMPLETE DESTRUCTION OF PLASTERING

No kind of plastering has been found to stand the heat of such fire as occurred in this building. Even the plastering that remained upon the wall would not have been strong enough to repair with additional coats.

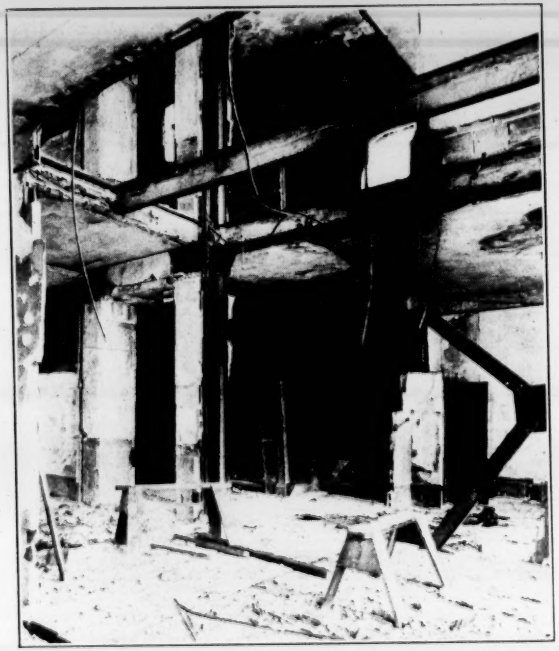

FIG. 9 LARGE HOLES IN FLOORS

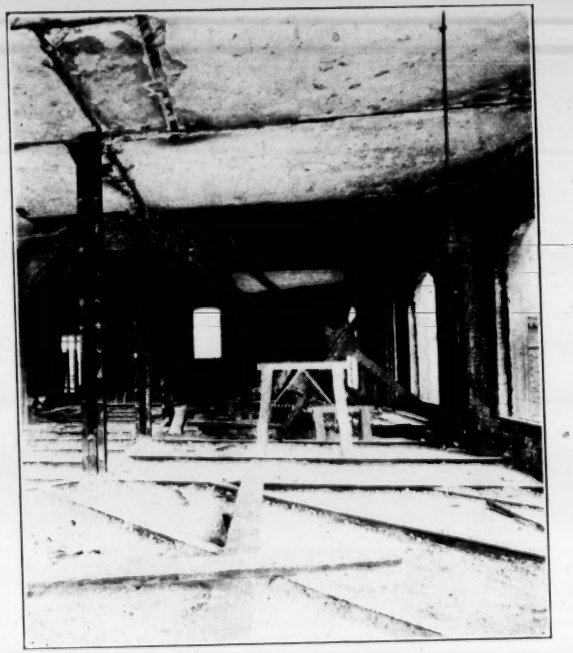

FIG. 10 VIEW OF SIXTH STORY

The large opening between floors (Fig. 9) was caused by structural steel in the roof falling through to the basement.

Fig. 10 is a typical interior view after partitions, cinder concrete fill, and all rubbish had been removed. The bricks in the walls were practically uninjured. The cinder concrete had no strength whatever. The hollow terra cotta floor tile slumped continually, necessitating walking on plank and beams instead of on the floor arches.

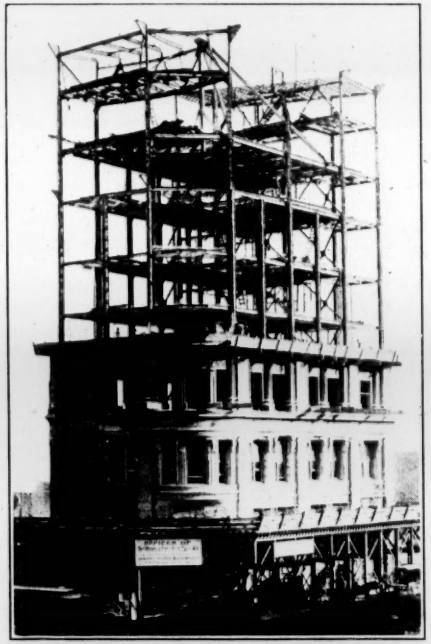

FIG. 11 NORTH ELEVATION

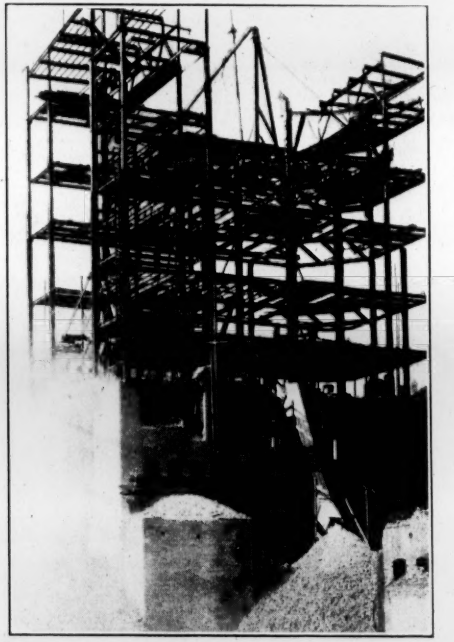

FIG. 12 EAST ELEVATION

Steel frame stripped to fourth story. Adjoining two different columns the entire height of the building (Fig. 11), connections had to be rebolted as fast as the work was taken down, asthe bolts were sheared by expansion of the brick walls.

The pile at the left (Fig. 12) shows brick that were sold and used again in new buildings. The rubbish pile of broken bricks, mortar, ornamental and structural terra cotta, cinder filling, etc., was about equal in size to the salvage. The steel below the eighth floor was uninjured. It was sold and used in other buildings in San Francisco.

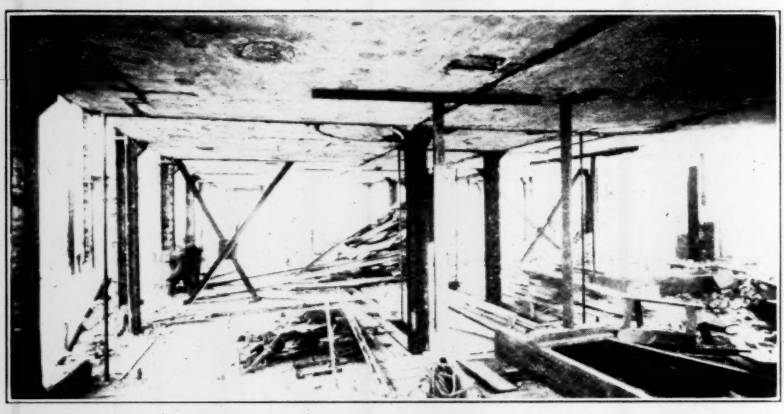

FIG. 13 DESTRUCTION OF FLOORS

The small holes in the ceiling were caused by workmen slumping through. The floors were so completely ruined by the heat that it was necessary to cover the third floor with plank to the depth of 18 in., to prevent accidents to the people in the lower stories from the slumping of the floors in the upper stories

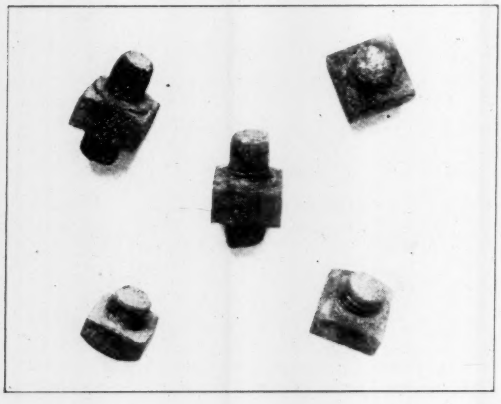

FIG. 14 SAMPLES OF BOLTS SHEARED BY EXPANSION DUE TO HEAT

FIG. 15 CONDITION OF STEEL FRAME

The left side of cut shows a typical exterior floor and column connection. The steel bears no signs of injury or disintegration, although buried in the exterior wall fourteen years. Fire had no effect on the steel, except warping where the frame was not properly protected by fireproofing, but there were several small places, such as that shown on right side of cut, that showed bad rust spots. Each of these cases appeared in steel buried in exterior walls. The rust spots appeared to be places that were not properly cleaned from rust at the time the columns were painted, and were imperfectly surrounded by brick mortar.

THE MECHANICAL ENGINEER AND THE TEXTILE INDUSTRY

By H. L. Gantt, New York

Member of the Society

The textile industry enjoys the distinction, to a greater extent, perhaps, than any other, of having been brought to a high state of perfection without the aid of the mechanical engineer. The machinery was developed by the mechanic before the mechanical engineer became a very important factor in the industrial world, and the plans were, and are still, built by mill architects, who as their name implies are architects rather than engineers. The most important field of this industry that the engineer has entered, is the power department, and in this field he has done much good work. The complicated and delicate machines for working cotton fibre, however, which are wonders of mechanical skill, have been brought to their high state of perfection by men who were mechanics rather than mechanical engineers. The operation of these machines, until recently, has been directed by men whose training was exclusively that of the factory, and who could solve well a concrete problem.

2 In this industry there is, as a rule, a wider gap between the financial interests that control, and the "help" that operate, than there is in almost any other industry.

3 The textile schools are today doing much to fill this gap by supplying to the mills educated men who, while understanding the detail operation of the machines, are capable of comprehending the larger problems of management, and can thus form a link between the financial men that control and the mechanics that operate.

4 The lack of such men in the past is undoubtedly responsible for the fact that some of the processes which influence the subject of management more than they do the product, and which are easily susceptible of being standardized and done automatically, are

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still being done expensively and inefficiently by hand, and in a manner that causes much subsequent labor and expense that should be avoided. The solution of this problem belongs more particularly to the function of management, for the workman often does not see the influence of one process on a subsequent one in another department.

5 I refer, for example, to the process of handling cloth in a bleachery. In order to make clear the point in mind, it is necessary to explain that in the process of bleaching, cotton cloth is generally sewed together piece by piece and handled in the form of a rope, which is drawn from one operation to the next by means of rolls. This rope of cloth is subjected to the action of various liquids, being first boiled in an alkali and then washed. After being washed it is usually impregnated with acid (technically "soured"), and allowed to stand in a pile for some minutes to allow the acid to act. The methods of forming this pile and of withdrawing the cloth from the pile, are the operations to which I have special reference.

6 As the piling operation is repeated after each of several impregnating operations, the successive pilings divide the process into a series of separate and distinct stages with a loss of time between every two. The usual method of piling is as follows:

7 The cloth is drawn from the souring machine by an overhead roll, which drops it to the floor beneath. A boy stands on the pile of cloth and so guides it with a stick that it is piled in substantially uniform horizontal layers. When the pile has reached a size determined by the judgment of the bleacher (or the boy), the rope of cloth is broken at a seam and a second pile is formed. When in the judgment of the bleacher the first pile has stood long enough, the cloth is withdrawn and pulled through a washing machine into a bath of chlorine water (technically "chemic"), after which it is again piled in the same manner by a boy with a stick. The judgment of the bleacher as to the time cloth should lie in a pile after impregnation seems to be controlled by his temperament, or by tradition, rather than by knowledge, for we find that hardly any two bleachers have the same opinion as to how long the cloth should be subjected to the action of the acid; and the practice varies from a few minutes to twenty-four hours. As a matter of fact the acid does all its work in ten minutes or less, and no beneficial effect can be discovered by a longer treatment.

8 Inasmuch as it is necessary to pull the cloth from the top of a pile, the leading portion as it leaves the sour pile has been acted

upon by the acid a shorter time than that at the bottom of the pile.

9 The top of the second pile is attached to the bottom strand of the first pile, and the top of the third pile is attached to the bottom of the second.

10 As each strand of cloth usually goes through several pilings in the course of being bleached, the action of the bleaching liquors on any portion of the cloth would be alternately long and short, according as that portion of the cloth was at the bottom or the top of a pile. If the rope of cloth was always broken in the same place, the worst that could happen would be an unevenness in the bleach due to the difference in treatment. It frequently happens however (and this is more often the case than not), that the rope of cloth is not broken in the same place; and when this occurs the various lots of cloth of which the rope is composed, which usually belong to different customers become almost hopelessly mixed. The expense of straightening out such a mix-up has usually been considered one of the legitimate expenses of bleaching. Add to this the fact that the piling boy often piles the cloth so carelessly that it tangles as it is pulled off the pile, and not only damages itself, but usually shuts down a portion of the plant for awhile.

11 If we also realize the fact that chlorine, or "chemic," not only forms a most unpleasant atmosphere to work in, but is actually injurious to the lungs, it would seem that some automatic piling machine which would hold the required amount of cloth and permit the leading end of the pile to be withdrawn would long ago have been devised. Inasmuch, however, as this is not a problem requiring great mechanical skill, but one requiring a somewhat different kind of knowledge, it apparently had never been attacked until the writer came in contact with it.

12 Fig. 1 shows the machine which has been developed to accomplish the result, and Fig. 2 shows the cloth as it is delivered to and withdrawn from the machine. The machine consists of an inclined chute, with upturned ends, and having a bottom composed of a series of independent rollers, freely revolving. The cloth is dropped into the tall stack, and falling on the rollers is carried by its own weight to the bottom of the incline. The incline is filled, and as the fabric rises in the receiving stack, the forward end of the pile is forced upward in the other end of the machine, from which it is pulled off at the rate at which it enters the receiving stack.

13 By making the chute of the proper length a pile of cloth of any size may be held, and the cloth may be subjected to the action of the

impregnating liquor for any desired time, all portions of the fabric receiving exactly the same treatment. Such action produces uni-

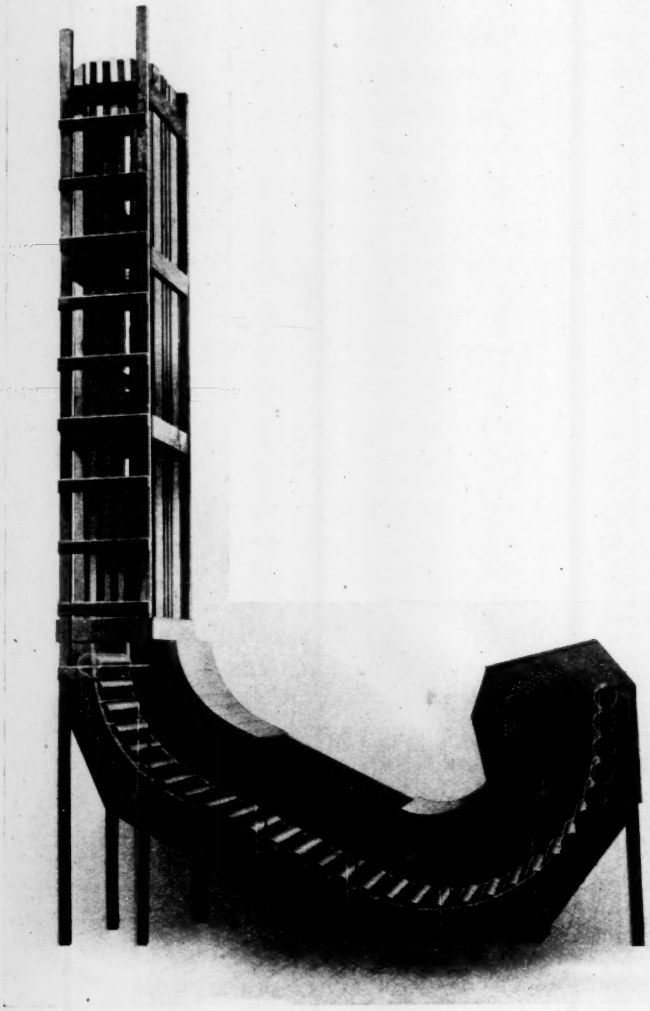

FIG. 1 AUTOMATIC PILING MACHINE

formity of bleach impossible under the old conditions, and as there is no need for breaking seams, the goods go through the bleach house in the order they went in, which produces a saving of expense and

worry realized only by the man who has operated under both methods. The straightening out of "mix-ups" and the "closing out" of "short lots" are the bane of a finisher's existence, and anything that reduces these troubles does much, not only to smooth the operation of the works, but to assure the customer that he is

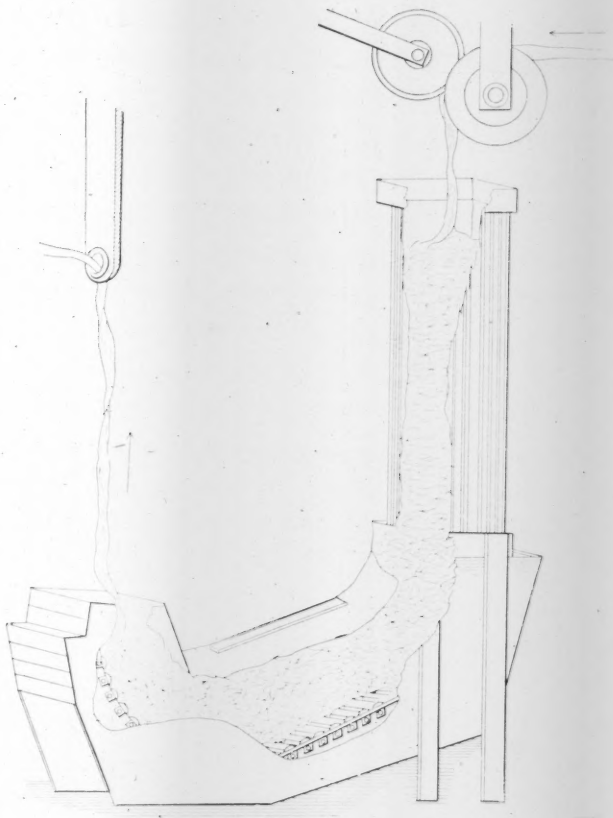

FIG. 2 PROGRESS OF CLOTH THROUGH MACHINE.

getting back exactly the goods he sent. Moreover the dirt and damage caused by the piling boys are eliminated.

14 The saving in always having clean goods in uniform condition is greater than the saving in wages of the boys, and the relief to the foreman of having a smaller number of bleach-house boys to manage, makes it possible for him to devote his time to bleaching rather than to boys, with distinctly beneficial results to the bleaching.

15 In addition to the advantages already mentioned, there is a marked saving in time, for the cloth remains subject to the action of each liquid only the time needed to produce the desired result. Each piling machine takes the place of from three to four bins, and as it takes up less space than one bin, the saving in buildings is very con-

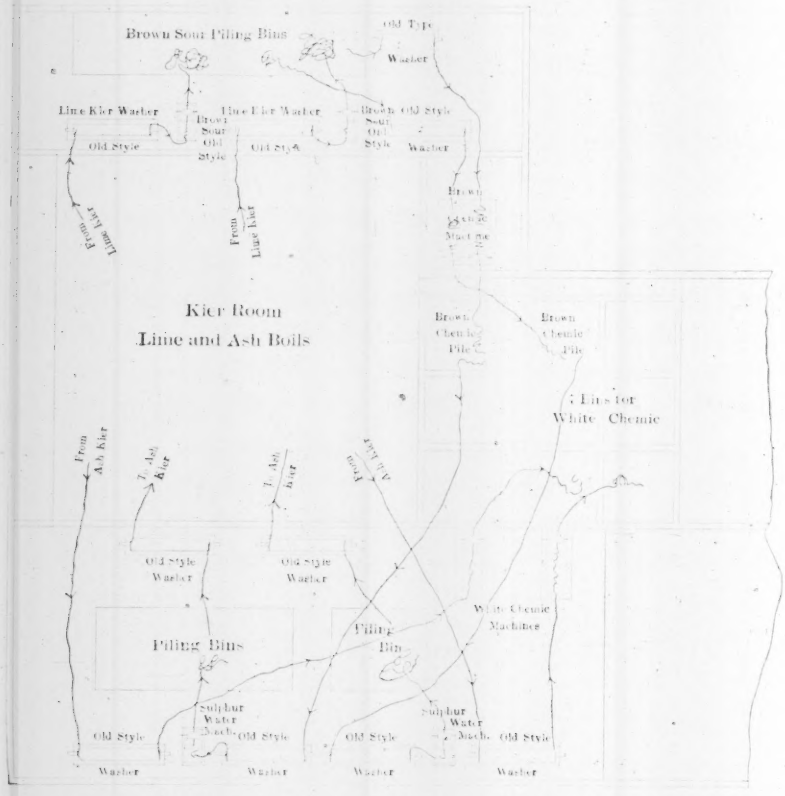

FIG. 3 ARRANGEMENT OF BLEACHERY (OLD SYSTEM).

siderable. In one case, when the washing, "souring" and "chemic" machines were rearranged in such a manner as to use a full equipment of piling-machines to the best advantage, the saving in bleach-house space amounted to more than 40 per cent. Wherever the machines have once been installed it is obvious that they soon become indispensable.

16 The fact that such an important operation can be taken care of in such a simple manner, is the best evidence that the writer entered

a field that has not been thoroughly investigated by the mechanical engineer. The field is still open, for plants are being built today to handle cloth exactly as it has been handled for fifty years. The builders of these plants have not yet discovered the function of the mechanical engineer, and are still putting their faith exclusively in bleachers.

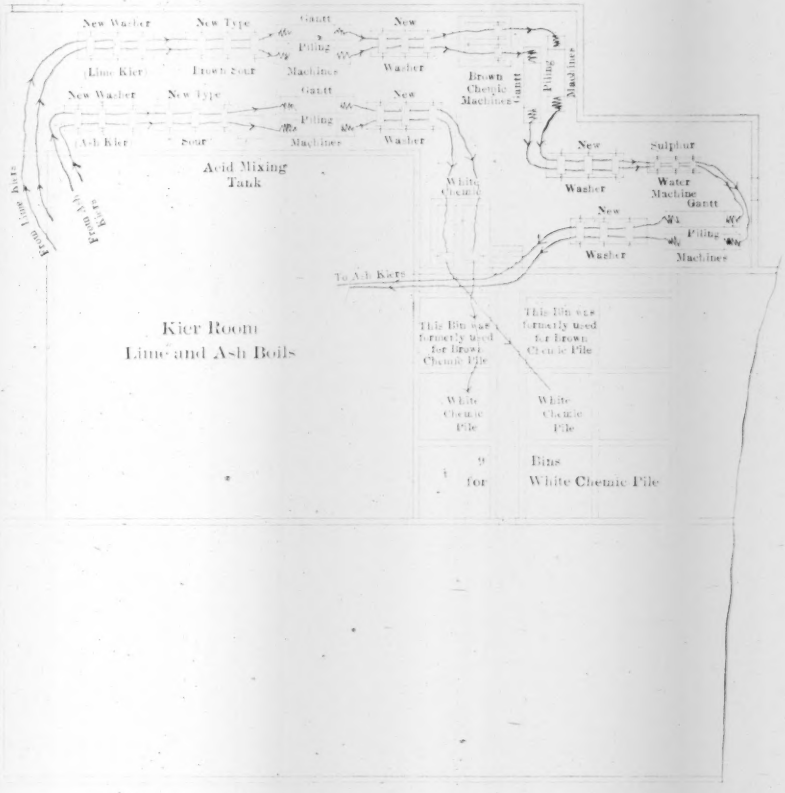

FIG. 4 ARRANGEMENT OF BLEACHERY (NEW SYSTEM).

17 Fig. 3 represents the course of cloth through a bleachery where the writer was told that the process had not been changed for fifty years. Fig. 4 shows the course of the cloth in the same bleachery after it had been equipped with piling machines, and other machines adapted to work economically with them. The new installation takes up less than 60 per cent of the space of the old and is operated by six people against the twenty people formerly needed.

18 The operations and their sequence are the same in the new as in the old lay-out, except for the installation of an additional "sour", which was thought desirable to remove the alkali of the second boil. The new lay-out is not a new method of bleaching, but simply a mechanical engineer's method of handling the old. There is hardly a bleachery in the country where the mechanical engineer cannot do similar work, and that without doing violence to the prejudices of the bleacher.

19 The standardization of bleaching methods must come later, and will take time, for we have here the habits of at least half a century to combat.

BALL-BEARING LINESHAFT HANGERS

By Henry Hess, Philadelphia, Pa.
Member of the Society

This paper is presented because of a request for more specific information made during the discussion of a paper that I was privileged to read at the Annual Meeting in New York, December 1909. That paper dealt with Lineshaft Efficiency, Mechanical and Economic, and was restricted to the presentation of actual measurements of the power consumption of the same lineshaft when mounted on plain and again on ball bearings, with that change the only variable.

2 It was the purpose of the paper to describe merely the actual test and its results, thus adding to the fund of available engineering knowledge. The favorable economic showing of the ball-bearing is inherent in the advantages of rolling as compared with sliding friction, quite aside from any particular make of ball-bearing; though if durability as well is to be secured, it naturally is essential that bearings of correct design and suitable workmanship and material be selected. The demand at the time of the discussion and since must serve as apology for such references to a product for which I am responsible as are incident to a description of the installation in question.

3 The first real improvement in hangers was made in this country by one of the oldest and most honored members of our Society. It was Mr. Bancroft who first mounted the box to swivel fully while well supported and who also gave it vertical adjustability. The means are so familiar to every one today as to constitute a commonplace. This original hanger was supplied with carefully fitted adjusting screws of liberal size; their ends terminated in correctly machined spherical sockets fitting over correspondingly machined spherical segments on the top and bottom of the box. Once adjusted in the hanger the box retained its place until a readjustment was demanded by some outside cause such as a settling building.

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4 Unfortunately the constant and incessant demand for lower and yet lower costs was given heed to by many until in course of time not a few of the many makes of hangers marketed are not machined at all but are foundry jobs throughout. Even so the consequent looseness and indefiniteness of position of the box supporting screws is not a very serious matter with plain bearings. The very length of the plain

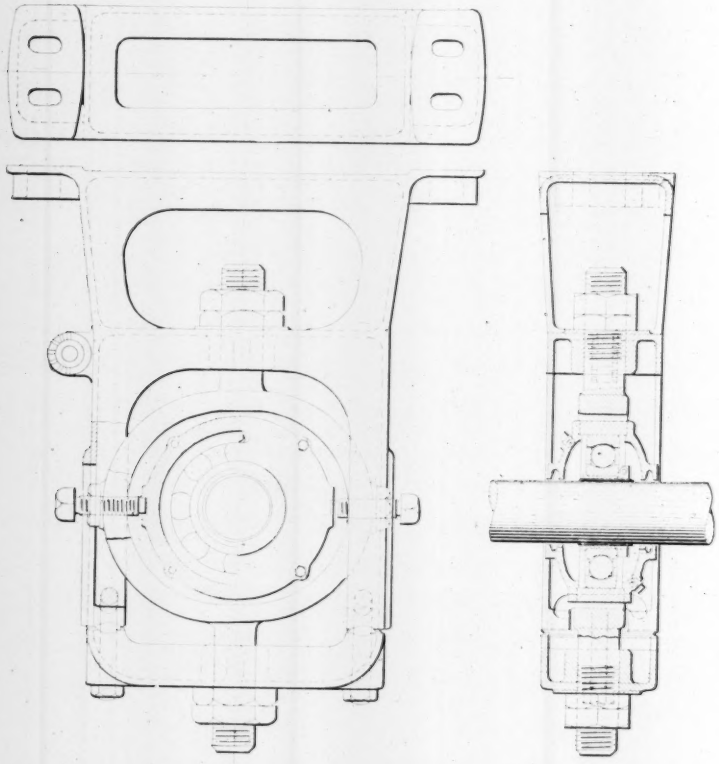

FIG. 1 ARRANGEMENT OF BALL-BEARING HANGER

bearing equal to some four or five shaft diameters helps to hold the box in line with the shaft if that is straight; or if the deflecting forces are too great, then the resultant excessive pressure on the plain bearings will draw attention to the trouble and by an insistent squeal, demand correction.

5 In first applying ball bearings to lineshafts I employed variou

hangers such as were on the market. The ball bearing was set in an enclosing box which in turn was held between the same supporting screws that supported the plain bearing displaced. Occasional reports were received that the power consumption was greater after the change than before. These were particularly insistent from a few installations where accurate meter records of electrical power consumed by the shaft driving motors were kept. Investigation finally disclosed in each instance a shifting of the shaft sections from their original alignment. The consequent and in several cases quite severe binding of the shaft at each revolution consumed considerable power. Owing to the very low coefficient of ball bearing friction, 0.0015, however, the additional load on the ball bearings was without evident effect and caused neither squealing nor even perceptible heating. Realigning the boxes immediately brought down the power consumption to the original low figure.

6 The Bancroft, or as they are better known, "Sellers" type hangers with their definitely machined supporting screws and sections of spheres on the outside of the box to rock on, was not found practicable with ball bearings, because these spherical sections became too flat with the large diameter of the box containing the ball bearing, as is evident from an inspection of the cross section of such a box given in Fig. 1.

7 It was finally and reluctantly decided to make up and build hangers, general coöperation with the specialists in that line having been found impossible from lack of interest, reluctance to make changes, the usual inertia resisting innovations, etc.

8 Many designs were laid out on paper and some tried, but again discarded as not responding to all of the following requirements of an ideal hanger:

a Definite support of the box on machined seats permitting no shifting under load.

b Ability of the box to swivel in all directions.

c Vertical adjustability of the box within the hanger body.

d Horizontal adjustability of the box within the hanger body.

e Rigidity in every direction, to permit the hanger to be used as a ceiling hanger, floor stand or post hanger.

f Convenience of adjustment in aligning the shaft.

g Adaptability for countershaft hangers involving a shifter-arm.

h Neatness of general outline and conformity to modern machine design by substituting box sections for ribbing.

i The lowest cost consistent with the other requirements.

9 The design finally evolved is shown in Fig. 1. The box is supported and pivoted horizontally on the ends of two screws tapped through the opposing sides of a yoke surrounding the box with enough freedom to give 1 in. of horizontal adjustment. This yoke has an upper and lower cylindrical stem, each turned to a running fit in the corresponding bore of the hanger; these stems are threaded and provided with nuts and checknuts and allow a vertical adjustment of 2 in. As the box can swivel horizontally in the yoke and that in turn vertically in the hanger the two together provide the desired universal swivelling freedom.

10 The hanger body is of channel section with the flat outside. The central crossbar that takes the weight of the box and shaft also materially stiffens the hanger. The lower crossbar is cast integrally with the hanger body and split off in the usual way. A single bolt at each end serves for attachment. This bolt is a drop forged T with the T turned down and ball ended to fit in corresponding sockets.

11 All of the various sizes, with the exception of the largest, which never comes into consideration for countershafts, are provided with cast-on lug with a serrated side face. To this may be bolted an arm of suitable length at any desired angle, to carry the usual eye guiding the shifter rod.

12 The construction of the bearing box is also apparent from Fig. 1, which shows it to consist of a central cast supporting ring bored to a sucking fit for the outer race of the ball bearing. To the side faces coverplates are bolted with an oil-tight joint and where these surround the shaft they are provided with a cored annular groove. The side lips are bored $\frac{1}{64}$ in. larger than the shaft diameter and have sharp edges.

13 This arrangement is found to be efficient in retaining the lubricant and so promotes cleanliness, while also preventing the entrance of ordinary foreign matter. For particularly difficult locations, as in cement mills, the very fine floating grit is kept out by a double groove, the outer one of which is filled with a fairly heavy grease. A single charge of heavy cylinder oil or non-acid grease will last the average lineshaft bearing several years; and to avoid total neglect refilling once a year is recommended.

14 The ball bearings are free endwise in the box. It is recommended that a lineshaft be held endwise by ordinary collars on either side of the central hanger so that all weaving endwise, due to expansion and contraction, settling, etc., is allowed for by the end freedom in the boxes. An occasional installation is found with heavy end-thrust in which case the thrust is taken on one of the ball bearings by letting

it come up against the shoulders of the coverplates. Unusually heavy thrusts are provided for by special hangers into which a collar type of ball bearing is built.

15 As it is necessary to clamp the ball bearing to the shaft so that the inner race cannot rotate and as commercial shafting is somewhat indeterminate in size a so-called "adapter" is employed, the details of which are shown in Fig. 2. The adapter consists of a bush fitting into the bore of the inner race, which in turn is fitted with a coned split sleeve. This latter is driven home endwise until the whole is tightly

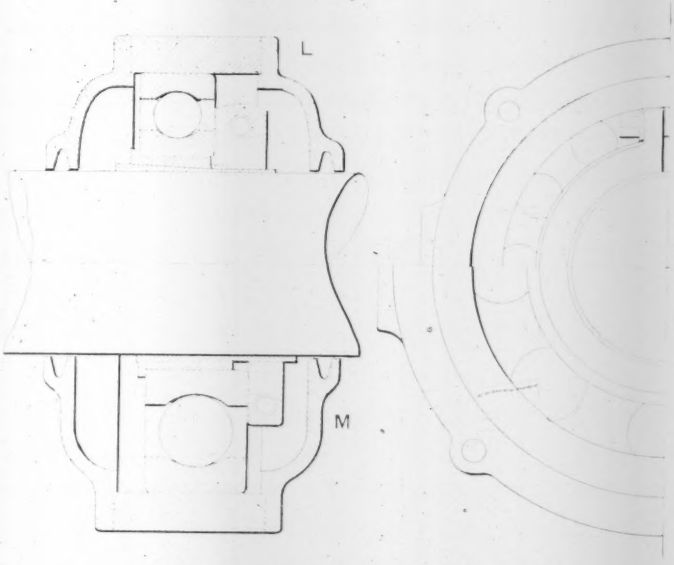

FIG. 2 BALL-BEARING ADAPTER

clamped onto the shaft retaining a truly concentric setting. Unless there is considerable vibration, the wedging effect of the small angle of the conical bush is sufficient to prevent any loosening, but to guard against all contingencies a split collar is clamped onto the sleeve with its face close up to the face of the bearing.

16 In the sectional view, Fig. 2, that part of the bearing above the center line appears narrower than that below, the first marked "L" and the latter "M." This indicates the relative widths of light-weight and heavy-weight bearings. Bearings of the "L" or light-

weight series take less load than those of the "M," or medium-weight, series. For all ordinary lineshafts the light-weight series is quite sufficient; it is only for extraordinary conditions or for jackshafts, etc., that the medium series comes into consideration. A single medium bearing is sometimes also used in the hanger next to the main belt.

17 The ball bearing itself is a very simple element, consisting, as is clear from both illustrations, of an inner race, an outer race, and a single row of interposed balls, running in grooves whose radius of curvature is only a few percent larger than the ball radius. Races and balls are made of alloy steels (a special mixture of the carbon, chrome, manganese variety).

18 These ball bearings are the same that are regularly supplied for the machine industries in general. Adapters for shafting range from $\frac{15}{16}$ in. to 3 $\frac{15}{16}$ in., both light and medium. As larger shafts are rarely of the nature of lineshafts, it is preferable to fit seats directly to the bearing bore without the interposition of adapters.

19 In the paper referred to at the outset direct measurements were cited showing a saving in lineshaft friction of about 35 per cent under proper belt tensions of 44 lb. to 57 lb. per inch width of single belt and correspondingly more for heavier loads. The return on the investment was also shown to be 37 per cent per annum, taking into account the higher cost of the ball-bearing installation. Such higher cost is, however, not necessary. The initial cost may, in fact, be actually lower if in the original layout full advantage is taken of the possibilities of the ball bearing. The ball bearing is as safe and reliable at 600 r.p.m. as at 200; whereas, the use of such high speed with plain bearings is beset with so many difficulties in the way of reduction of size of lineshaft, pulleys, drop of hangers, etc., as to take that practically out of consideration for the average plant. The first cost of a ball bearing installation of 600 r.p.m. compares favorably with the first cost of one at 200 r.p.m. on plain bearings and the full advantage of the saving to be derived from ball bearings may thus be realized without any extra investment whatsoever.

20 In fact, the first cost of the ball-bearing equipment is quite likely to be less. As an example, it is fair to take a lineshaft such as that on which the test referred to in the December paper was made, but fully loaded to drive 32 machines. The installation elements that vary with the shaft speed are the shaft diameters, pulley diameters, hanger drops, belt widths and countershaft-driven pulleys.

The net cost of the installation material for the slow speed shaft of 200 r.p.m. on plain bearings transmitting 40 h.p. is	$558.95
The net cost of the alternate installation for the high speed shaft of 600 r.p.m. on ball bearings is	441.87
The saving in first cost of material amounting to 21 per cent	$117.08

See Appendix for details.

21 The lineshaft bearing friction saving, based on an average coefficient of friction and confirmed by the series of tests cited in the previous paper, is 0.41 kw. or 1230 kw.-hr. for a year of 3000 working hours and worth, at 3 cents per kw.-hr., $36.90; secured by an installation 21 per cent lower in initial cost. See the Appendix for further details.

APPENDIX

COST OF PLAIN-BEARING SLOW-SPEED LINESHAFT MATERIAL

Shaft 200 r.p.m. in ring-oiling boxes, transmitting 40 h.p., driving 32 countershafts, one belt to each countershaft, 8 ft. to countershaft centers, countershafts 300 r.p.m.

10 hangers, 16-in. drop, ring-oiling babbitted boxes, net cost			$53.63
1 main pulley, 40 -in. x $12\frac{1}{2}$-in. double belt			
4 line pulleys 15 -in. x $6\frac{3}{4}$-in. single belt	4 counterpulleys	10 -in. x $6\frac{3}{4}$-in.	
4 " " $8\frac{3}{4}$-in. x $4\frac{1}{2}$-in. single belt	4 counterpulleys	6 -in. x $4\frac{1}{2}$-in.	
4 " " 10 -in. x $4\frac{3}{4}$-in. double belt	4 counterpulleys	$6\frac{3}{4}$-in. x $4\frac{3}{4}$-in.	
4 " " $9\frac{7}{8}$-in. x $5\frac{1}{2}$-in. single belt	4 counterpulleys	$6\frac{1}{2}$-in. x $5\frac{1}{2}$-in.	
4 " " $9\frac{3}{4}$-in. x $7\frac{3}{4}$-in. double belt	4 counterpulleys	$6\frac{1}{2}$-in. x $7\frac{3}{4}$-in.	
4 " " $8\frac{1}{4}$-in. x $4\frac{1}{2}$-in. single belt	4 counterpulleys	6 -in. x $4\frac{1}{2}$-in.	
4 " " 10 -in. x $4\frac{3}{4}$-in. double belt	4 counterpulleys	7 -in. x $4\frac{3}{4}$-in.	
4 " " $12\frac{1}{4}$-in. x $4\frac{1}{2}$-in. single belt	4 counterpulleys	8 -in. x $4\frac{1}{2}$-in.	
		Net cost	117.97
20-ft. 12-in. double belt			
320-ft. 3-in. single belt			
160-ft. 3-in. double belt			
80-ft. $3\frac{3}{4}$-in. single belt			
80-ft. 4-in. double belt			
		Net cost	360.10
72-ft. $2\frac{7}{16}$-in. lineshaft		Net cost	27.25
		Total net cost	$558.95

COST OF BALL BEARING HIGH SPEED LINESHAFT MATERIAL

Shaft 200 r.p.m. in Hess-Bright medium series ball bearings, transmitting 40 h.p., driving 32 countershafts, one belt to each countershaft, 8 ft. to countershaft centers, countershafts 300 r.p.m. Lineshaft pulleys reduced in diameter by one-half and the driven countershaft pulleys incerased in diameter by one-third; belts decreased in width in correspondence with the increased speed; lineshaft decreased in diameter proportionately to speed; under both conditions safe for 60 h.p. according to catalog ratings. Hanger decreased in drop to 10 in., rather more than the diameter of largest counterdriving pulley. Belts 50-lb. per inch width, single.

10 hangers, 10-in. drop, medium weight ball bearings, net cost		$159.20
1 main pulley, 28-in. x $6\frac{1}{2}$-in. double belt		
4 line pulleys $7\frac{1}{2}$-in. x $4\frac{1}{2}$-in. double belt	4 counterpulleys 15-in. x $4\frac{1}{2}$-in.	
4 line pulleys 6-in. x $3\frac{1}{2}$-in. single belt	4 counterpulleys 12-in. x $3\frac{1}{2}$-in.	
4 line pulleys 5-in. x $2\frac{1}{2}$-in. single belt	4 counterpulleys 10-in. x $2\frac{1}{2}$-in.	
4 line pulleys 5-in. x $2\frac{3}{4}$-in. single belt	4 counterpulleys 10-in. x $2\frac{3}{4}$-in.	
4 line pulleys 5-in. x $4\frac{1}{2}$-in. double belt	4 counterpulleys 10-in. x $4\frac{1}{2}$-in.	
4 line pulleys 6-in. x 2 -in. single belt	4 counterpulleys 12-in. x 2 -in.	
4 line pulleys 5-in. x 3 -in. single belt	4 counterpulleys 10-in. x 3 -in.	
4 line pulleys 8-in. x 2 -in. single belt	4 counterpulleys 16-in. x 2 -in.	
	Net cost	87.57
20-ft. 6 -in. double belt		
240-ft. 2 -in. single belt		
240-ft. $1\frac{1}{2}$-in. single belt		
80-ft. $2\frac{1}{2}$-in. single belt		
80-ft. $2\frac{3}{4}$-in. single belt		
	Net cost	162.00
72-ft. $1\frac{11}{16}$-in. lineshaft	Net cost	13.10
	Total net cost	$441.87

ESTIMATE OF SAVING IN KILOWATTS

In the Author's paper, on Lineshaft Efficiency, published in The Journal for December 1909, Par. 32, is given: $Kw = 0.0000059\ Ldsμ$

		Plain Bearings	Ball Bearings
L	= total journal load in lb. =	9606	5420
d	= shaft diameter in inches =	$2\frac{7}{16}$-in.	$1\frac{11}{16}$-in.
s	= shaft speed in r.p.m. =	200	600
$μ$	= coefficient of friction =	0.03	0.0015
Kw	= kilowatt friction loss =	0.90	0.49

Kw., plain $= 0.0000059 \times 9606 \times 2\frac{7}{16} \times 200 \times 0.03 = 0.90$

Kw., ball bearing $= 0.0000059 \times 5420 \times 1\frac{11}{16} \times 600 \times 0.0015 = 0.49$

Kw. saving $= 0.90 - 0.49 = 0.41$

THE HYDROSTATIC CHORD

WITH DISCUSSION OF ITS APPLICATION IN THE DESIGN OF LARGE PIPES OF REINFORCED CONCRETE

By Raymond D. Johnson,[1] Niagara Falls, N. Y.

Non-Member

The hydrostatic chord is allied to the catenary, the parabola and the circle, because all of these curves may be formed by a flexible inextensible substance, supported at its two extremities and properly loaded. If the load is uniformly distributed with respect to a horizontal line joining the supports, the action of gravity will shape the supporting substance to form a parabola. If the load is uniformly distributed with respect to the curve itself a catenary is the result. If the load is applied by fluid pressure, irrespective of the direction of gravity, so that the pressure is of uniform intensity normal to the curve, a circle is formed. If the load is applied by fluid pressure which varies according to the head or depth of water at any point, the curve resulting from this system of forces normal to the curve is a hydrostatic chord, which can easily be imagined as the curve which a flat canvas hammock would take if filled with water.

2 If the canvas were sewed together to form a closed curve, and supported on end as a vertical cylinder, the cross section would become circular under fluid pressure. If now the open ends of the cylinder were sealed with flexible bulkheads and the cylinder was tipped over on its side when completely filled with water the cross section would become a hydrostatic chord, although it would still theoretically be a circle also, until a drop of water was allowed to escape, that is, if the water be regarded as incompressible. Since the shell of the cylinder is assumed inextensible, and a circle encloses the maximum area for a given perimeter, it follows that the mere act of tipping such a cylinder towards the horizontal position would immediately develop infinite stress in the enclosing membrane, if the water could not par-

[1] Ontario Power Company, Niagara Falls, N. Y.

The American Society of Mechanical Engineers, 29 West 39th Street, New York. All papers are subject to revision.

tially escape. If one now imagines a hole pricked in the side of the cylinder on top, and connected with a vertical pipe or piezometer, some of the water would escape up into the pipe and at the same time the shape of the cylinder would take on the characteristics of a true hydrostatic chord. The water would continue to rise until the head above the top was just sufficient to hold the remaining water in the equilibrium shape. At this time the surrounding membrane would be in pure tension, equal throughout, and of finite value. The very fact that the tension must be constant in all portions of the shell furnishes an easy means of constructing its shape, because the tension at any point is obviously measured by the product of the radius or curvature and the head at that point.

3 If more water now be poured into the piezometer pipe, the shape of the cylinder will approach a circle; its vertical diameter will lengthen and its horizontal diameter will shorten. It would, however, never become a circle for a finite head of water. Conversely, if water be drawn off, the shape would become more and more oblate until finally the membrane would collapse into a plane surface at zero head.

4 It seems apparent from the above discussion that a circle is not the natural shape for a pressure pipe lying on its side, especially if its diameter is large as compared with the water pressure. No one would think of designing a vertical water tank with an elliptical cross section, and thus subjecting the shell to enormous and unnecessary deforming stresses in its effort to become circular. And yet it has not been unusual, not only to design concrete pressure pipes circular, but even to go to the other extreme and shape the section with its least radius of curvature at the top instead of at the haunches.

5 This latter procedure might be compared to designing a stiff suspension bridge cable, say, for the sake of illustration, of reinforced concrete, and shaping it like an ellipse with its long axis horizontal, instead of the more natural shape in which the radius of curvature would decrease toward the center, instead of increasing. Such a chord is obviously so ridiculous that an example of it could not be found in practice. Instead it would probably be designed of parabolic shape, which would be perhaps as good a compromise as one could reach. If it were really too stiff to adjust its shape to changes of load, as for example when a moving load passed over the bridge, then more or less severe deforming stresses would be the result, but how much less than in the former case.

6 Similarly, although it is impossible to design a stiff pipe of

such a shape that there will be no deforming stresses under the varying conditions of water pressure and back fill and the constant weight of the shell itself, yet these stresses can be reduced to a minimum by adopting a form which lies midway between the ideal equilibrium shapes which the pipe tries to assume under the various water pressures to which it may be subjected. This matter is of much practical importance in large conduits, proper design of the cross section meaning the saving of perhaps one-half the material, concrete or steel.

7 It is not the writer's purpose to take up the mathematics of the hydrostatic chord, nor to follow through the complications of a typical design. The element of judgment enters so largely into such a study that it is impossible to do it justice in restricted space and time. A few general hints may be of service.

8 It is well known that a circular cylindrical shell lying on its side has four nodes, or points of contra-flexure, due to its own weight. These lie at points 50 deg. 36 min. 45 sec., and 146 deg. 19 min. 25 sec., respectively, from the vertical. It can be demonstrated that the locations of the nodes due to the weight of the water within such a cylinder are the same. It can also be shown that the bending moments due to both causes are exactly proportional at all points of the arc, and may therefore easily be combined. The equilibrium shape which would sustain the existing water pressure without any tendency to deform, can easily be plotted from the polar equation of the bending moments in a circle, in terms of the angle of departure from the vertical, remembering that the radial intercept between the circle and the new curve at any point, is a measure of the bending moment at that point, and when multiplied by the corresponding tension at that point of the circle will give the value of the bending moment.

9 Conversely, if the bending moment be divided by the tension, the radial intercept will be the quotient, and may be plotted. The value of this intercept at any angle, and for any assumed head H above the top of the pipe, is as follows:

$$\frac{r\,(\frac{1}{2} - y)}{H + \frac{r}{r}} - y$$

where r = radius of the circle, and $y = \frac{1}{4}\cos\phi + \frac{1}{2}\phi\sin\phi$.

10 Any number of such curves may be plotted, according to the number of different values of H assumed, and all of these will of course pass through the same node points.

11 These equilibrium shapes are not identical with the hydrostatic chord, for the reason that the forces acting to produce the latter are strictly normal to the curve itself, whereas in the former case the applied forces are always considered to be normal to the common circle for all the different equilibrium curves.

12 The discrepancy between the two curves for any given head, is, however, so slight as to be negligible for all practical purposes. A measurable difference would scarcely be found except for very low heads, say less than one diameter above the top of the pipe, where the stresses are small and not important. After the head reache

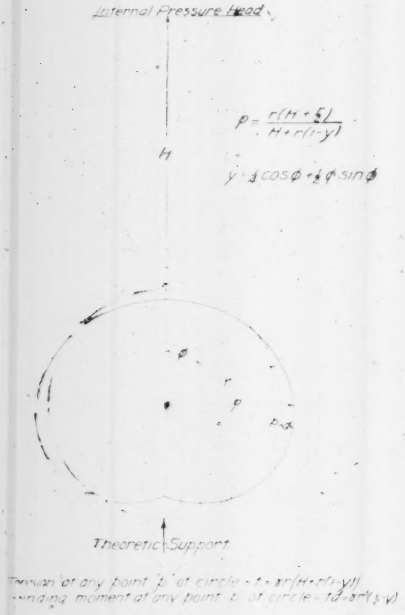

FIG. 1. EQUILIBRIUM CURVE

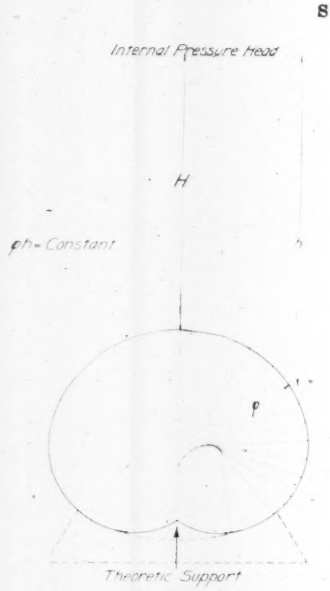

FIG. 2 HYDROSTATIC CHORD FOR SAME HEAD

a value of five or six diameters above the top of the pipe, the departure from a circular shape is comparatively slight, and therefore this discussion is particularly applicable only to large pipes under low pressure.

13 It will be found that when the pipe is supported on a continuous saddle, as is usually the case, the maximum stress is likely to be located at the top of the pipe, so that if the shell is made homogeneous no other point need be investigated for stress. If the pipe, in-

stead of being circular, is formed on the lines of any one of the equilibrium curves, or better still, on the lines of the true hydrostatic chord for a particular head, then the bending moments induced in it by any other head may be scaled or computated at any point by noting the length of the intercept between such a curve and the curve corresponding to the head under consideration. The value of the tension at any point, which must be multiplied by this intercept, is given by the formula $\alpha r\,[H + r\,(1 - y]$, where α is the weight of a cubic foot of water and the other quantities remain as before.

14 In arriving at the total stress it is necessary to combine algebraically the bending stresses due to weight of shell, back fill and weight of water, and then add to them the tensile stress due to the water pressure. The bending moments in a circle when lying on a flat surface are simply expressed by the equation, $-\alpha\, r^3\,(\frac{1}{2} - y)$.

15 When the pipe rests on a saddle the maximum stress is usually found at the top of the pipe, as before stated; and although its amount is somewhat lessened by the presence of the saddle underneath, it is not considered advisable to rely on this, and it is best to design for stress at this point strictly as though the pipe had a very narrow saddle, or theoretically none at all.

16 The above reasoning is not to be regarded as hard and fast for text-book use, and the conclusions are known to be merely close approximations. Many interesting properties of the hydrostatic chord have been studied by the writer, and much of the mechanics relating to pipe design has been more or less thoroughly worked out; but there is so much of no practical application, and so much time would be needed to coördinate the material, that no attempt has here been made to do so.

17 The curve itself is not new if one considers it the same as the hydrostatic arch with the stresses all reversed, but very little, if any, study of its properties and practical application has been published. Nothing of the kind has ever come to the notice of the writer. As a matter of theoretical mechanics and even of plane geometry there seems to be an opportunity for a little addition to the technical instruction in those branches.

THE SHOCKLESS JARRING MACHINE

In this paper the following subjects have been treated:

	Paragraphs
The term Shockless as a distinctive name for a new type of jarring machine	1-25
The jarring machine and its use defined	2
History and development of jarring and other molding machines	3-4-5-6-7-8
The value of labor saving appliances and the enormous increase in production made possible by their combination and concentration	9-10-11-12-13
The cost of savings and the expedients used to avoid subsequent damage	14
Object of this paper to elucidate the principles upon which the shockless jarring machine works and establish its superior claim to efficiency	15
The jarring process very quick and effective, although not always efficient in the consumption of power	16-17
Solidity of construction the most important consideration	18
Construction of table to minimize vibration and to provide solidity of mounting for patterns and flasks	19
Possibility of the unlimited consumption of power in jarring sand to a given density, and advantage of long stroke	20
Pressure of impact measured by change in velocity of table, and ramming effect measured by the square of this change in velocity at the instant of impact	21
Highest mechanical efficiency points to an anvil of infinite weight or to one founded on rock, but ground waves make this construction impracticable	21
Anvils cushioned on timber cribbing less effective	22-23
General description of the shockless jarring machine	25-26-28-29
The shockless anvil twice as efficient as the same anvil cushioned on a wooden crib	31-34
Change in velocity of table by impact as affected by condition of sand	33
No machine perfect from every point of view and the best machine the best compromise of conflicting advantages	34
Experiment demonstrating the feasibility of installing shockless jarring machines on the upper floors of high buildings	35

THE SHOCKLESS JARRING MACHINE

By Wilfred Lewis, Philadelphia, Pa.

Member of the Society

The title of this paper may appear to the casual reader like a contradiction of terms, but to the foundryman who has hitherto attempted to ram sand by the jarring process, the term "shockless" will be understood to apply only to the foundation or support on which the machine stands.

2 The jarring machine is essentially a sand-packing machine, capable of ramming any mold, large or small, in a minute or less time. By the method employed, the sand is rammed, as it should be, densest at the surface of the pattern and of decreasing density above, thus favoring the escape of gases when the mold is poured. The packing of the sand results from impact between the table on which the mold is carried and the anvil on which it drops. Various means may be used to lift the table and let it drop, but in foundry work compressed air has come to be generally preferred for its convenience as a medium for the transmission of power, and also for the simplicity of the machines resulting from its use. The jarring machine is not universal in its application, nor should it be used without judgment and discrimination. Due regard must be given to the construction of the pattern so as to permit a flow of sand chiefly in one direction, and to withstand successfully the shock of impact in ramming. But, for the broad field of work adapted to its use, there is nothing comparable to the jarring machine as a saver of time and money.

DEVELOPMENT OF MACHINE MOLDING

3 Jarring machines have been in practical use for many years without attracting much attention. The records of the patent office go back to 1869; but like all other types of molding machines, they

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have never been made, until quite recently, for large, heavy work beyond the strength of one or two men to handle. It was quite natural that the molding machine should begin its development in a small way on small work; as the field has widened it has been seen that the possibilities for saving time and money in the foundry have increased with the size of the work adapted to machines.

4 Anyone who has watched a bench molder fill his little flask, ram it in a few seconds with a butt rammer in each hand, then roll it over and draw the pattern with the dexterity of an artist in legerdemain, can appreciate the difficulty of helping him in his work by any mechanical means. Yet the old-fashioned hand squeezer saved some of the ramming time, and the power squeezer saved a little more, but not so much in time as in the strength of the operator, keeping him fresh, with steady hand and eye, for the delicate work of drawing patterns and setting cores. Pattern guides were then devised to assist in drawing patterns, and vibrators were invented to free the pattern from the sand without appreciably enlarging the mold.

5 The use of molding machines on small work has resulted in a substantial saving in the cost of molding, less wear and tear on patterns, and greater uniformity in castings, the saving in weight of the castings due to the use of a vibrator being sometimes an item that soon pays for the installation of the machine.

6 The hand squeezer is, of course, limited in its application to what can be done at one effort by one man, and for larger work power squeezers have generally been employed; but when large molds are squeezed by power, more or less trouble is encountered in the distribution of pressure on the sand. At one time an effort was made to overcome this difficulty by means of a water-bag placed between the sand and the squeezing head. Better results were obtained, however, by judicious tucking in deep pockets, by heaping the sand over deep places, and by scooping it out over high points in the pattern; but the main difficulty in squeezing deep molds lies in the fact that the sand is generally moved against the pattern instead of the pattern against the sand. This results in the greatest density of sand being away from the pattern, next to the squeezing head, and not where it should be, next to the pattern.

7 As shown by Harris Tabor's experiments,[1] presented to this Society in 1892, the friction of sand on the sides of a deep flask may carry a large part of the pressure on the squeezing head. To avoid

[1] Transactions, vol. 13, p. 537.

this difficulty, bottom-ramming machines have been employed, which move the pattern against the sand, but this method contemplates a definite, predetermined movement of the pattern to produce a mold of the density desired, and is subject to variations not easily controlled. Bottom ramming has, therefore, not been adopted to any great extent, and power squeezers have usually been limited in application to flasks not more than two to three feet on a side by a foot deep. Such machines, when designed also for pattern drawing, are comparatively expensive and have marked for a time the limitations of machine molding.

8 During recent years, however, while the power squeezer and the split-pattern machine were completing their development, the much neglected jarring machine has grown steadily in favor and in size, until today there would seem to be no limit to its capacity. These machines are simple in construction and effective in operation, while on large work the saving to be effected by their use probably exceeds that by all other types of molding machines combined. I say on large work, because on small work jarring machines cannot compete with power squeezers of the same capacity, except perhaps in a few special cases where the work is deep.

VALUE OF LABOR-SAVING APPLIANCES

9 The value of any machine depends of course on what it can save, and what it costs to effect that saving; a problem to be worked out in every instance by a systematic time study of all the operations embodied in producing a given result. For instance, if it takes two men eight hours to mold by hand a certain pattern, in a flask 45 in. by 60 in. by 36 in., and if five hours of this time is consumed in ramming sand, a jarring machine would save practically five hours of the time. It would not save any of the pattern-drawing and finishing time, nor any of the time for setting cores, but it would enable two men to make the mold in three hours, instead of in eight hours by hand. Hence, with a suitable jarring machine, two men could put up 2.67 times as much work as by hand.

10 In this case the jarring machine saves more than half of the molding time, and is therefore the most important help in the reduction of cost; but when patterns are rapped with a sledge, and drawn with a crane or hoist, a great deal of the molder's time may be taken up in finishing, or, to put it in more bluntly, in repairing the damage done to the mold by this brutal way of rapping and drawing the pat-

tern. Assuming that about one hour might be spent in finishing each half-mold when made by hand, an effective pattern-drawing machine could easily save two hours; and it is evident that with such a machine two men could make a mold in six hours, thus increasing their rate of molding 1.33 times that by hand. By means of a sand conveyor, or even a clam-shell bucket on a traveling crane, perhaps 30 minutes could be saved, thus enabling two men to make the mold in 7½ hours. With this device only, they could make their rate of molding 1.067 times as fast as by hand.

11 For the purpose of illustrating the effect of coöperation or concentrated effort upon any given piece of work, let us now assume that the demand for the castings above referred to has resulted in the making of three sets of patterns, and that we have three sets of men at work making three molds a day by hand. Now, suppose we give one set of men a jarring machine, another set a pattern-drawing machine, and the third set a sand conveyor. In 8 hours:

The first	set of men	will	produce		2.67	molds
" second	" "	"	"		1.33	"
" third	" "	"	"		1.067	"

Six men with three patterns will produce 5.067 molds, instead of three molds by hand, or about 1.7 times as much work.

12 On the other hand, if we have but one pattern and one set of men, and give them the combined help of a jarring machine, a pattern-drawing machine and a sand conveyor, two men will save 5 hours in ramming time, 2 hours in finishing time, and half an hour in shoveling sand, or 7½ hours in all; bringing the time on one mold down from 8 hours to 30 minutes, and increasing the production 16 times. In other words, the same assistance concentrated for the benefit of two men will result in more than three times the production, at less than one-ninth the cost per mold.

13 The above illustration shows, not only the advantage of concentrated effort in the use of labor-saving appliances, but also the wide difference in results that may be realized from the same appliances in different hands. Not only does the concentrated effort in this case save the wages of four men and produce three times as much work, but it also distributes all of the indirect charges, which must ultimately be carried by the product, over a larger output. So much for the savings to be effected.

14 On the other hand, the interest on the investment, the depreciation of the machine and the consumption of power, must be

accounted for; and in addition to all these, the damage that may be caused by the action of the jarring machine upon finished molds, or even upon buildings in the neighborhood, and the annoyance caused by noise and ground waves generally. This damage and annoyance has increased steadily with increase in the size of the machines and in the weight of the loaded table. To meet these serious objections, various expedients have been adopted, among which may be mentioned a reduction in the stroke or drop, and a more or less resilient bedding for the anvil.

PRINCIPLES GOVERNING THE DESIGN OF A JARRING MACHINE

15 These palliatives, however, left much to be desired until the shockless jarring machine, with its anvil rising up to meet the falling table, was developed. This has eliminated the chief objection to jarring machines. It is the object of this paper to elucidate the principles upon which it works and to establish its claim to superior efficiency in the consumption of power.

16 In the first place, it must be admitted, that although the packing of sand by the jarring process is very quick and effective in producing results, it is not very efficient under the most favorable conditions, as far as the expenditure of power is concerned, and that under certain conditions the efficiency may be reduced to zero. In the process of ramming, the density of sand is increased 25 or 30 per cent, and if a steam indicator were attached to the cylinder of a power squeezer, it may be questioned whether it would ever show over 1000 ft. lb. per cu. ft. as the work actually done on the sand in squeezing it to proper density. Of course a great deal more power than this would be consumed in the use of water or air as a working fluid, but the work put into the sand would in all probability not exceed 1000 ft. lb. per cu. ft.

17 To produce the same effect by jarring, the sand might be raised to a height of 4 in. and dropped upon an anvil 30 times; but to the weight of the sand must be added the weight of the table and flask, and the excess sand used as an aid in ramming. The first blow struck will cause the greatest flow of sand and will do the most work upon it, while each succeeding blow will increase the density and do less and less work, until, after a certain number of blows, the sand will remain at a density corresponding to the drop. When this point has been reached, the continued action of the jarring machine simply wastes power and produces no effect. The jarring machine is therefore

more efficient during the earlier stages of the process than it can be when the condition of maximum density is approached; and for this reason, the longer the stroke the greater will be the efficiency. Other considerations of a practical nature, affecting the elasticity or durability of flasks and patterns and of the machine itself, necessarily tend to limit the stroke, however, so that in practice it varies from $\frac{3}{8}$ in. on some machines to 4 in. or more on others with an average of perhaps $2\frac{1}{2}$ in.

18 In such machines, the most important consideration is solidity of construction and freedom from vibration of the jarring table. Otherwise the sand will become broken or laminated and the mold will be liable to fall apart in handling. Although lightness of construction in the jarring table is obviously desirable from the standpoint of economy in power, it is certainly not desirable from the standpoint of making perfect molds. The good results which accompany the stronger and stiffer table really cost less and it consumes less power in the end, because there are no failures necessitating repetition, or molds to be thrown away. The importance of solidity in the jarring table will be appreciated after a consideration of the character of rammed sand. It has a certain amount of elasticity, a good deal of resistance to further compression, and some tensile strength, which of course is easily overcome. There must, therefore, be no movement between the pattern, sand and flask, tending to pull the sand apart, and of sufficient amplitude to cause fracture, and no lateral movement tending to slide one layer of sand over another. Such fracture or lamination may be caused by badly fitted pattern boards, flimsy patterns, or crooked flasks not properly bedded, but a light and flimsy jarring table that can be easily warped out of shape will augment the difficulty, and effectually prevent the success of good patterns carefully mounted in strong and firmly bedded flasks.

19 In the molding machine to be described, the table adopted has been formed in one piece with the jarring cylinder, as shown in section in Fig. 2. This table has great depth of beam, and the metal is distributed as it should be for economy of cast iron, in a broad expanse of plate on the tension side, and a smaller mass around the cylinder on the compression side, where the blow is struck. Radial ribs connect the tension and compression sides of the beam, forming a table of enormous strength and stiffness to distribute the central blow of impact equally in all directions. A table of this type is really stiffer than some of the anvils on which other tables are made to drop. At any rate, there is no perceptible vibration of the table when it strikes

its anvil, or rather the buffer ring of leather, or other non-resilient material, interposed to relieve the sharpness of the blow and to reduce the noise. This buffer also helps to reduce vibration and rebound, by reducing the intensity of the force of impact. It is not, however, the rebound of the table from its anvil that injures a mold, so much as the rebound of the flask and sand from the table. Solidity of contact between table, pattern board and flask, is one of the most important elements for the successful working of a jarring machine; yet this detail rarely receives the attention it deserves, and not unfrequently the machine is condemned for this reason, which is no fault of its own.

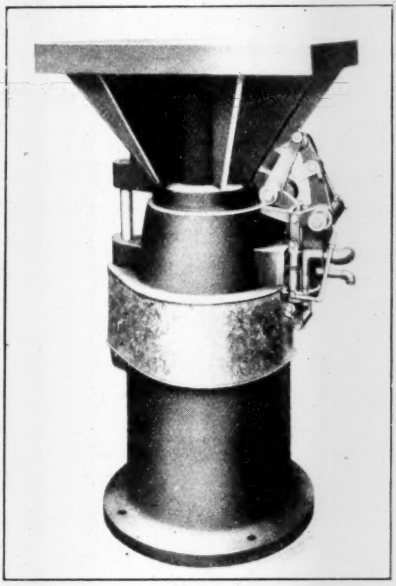

FIG. 1 SHOCKLESS JARRING MACHINE

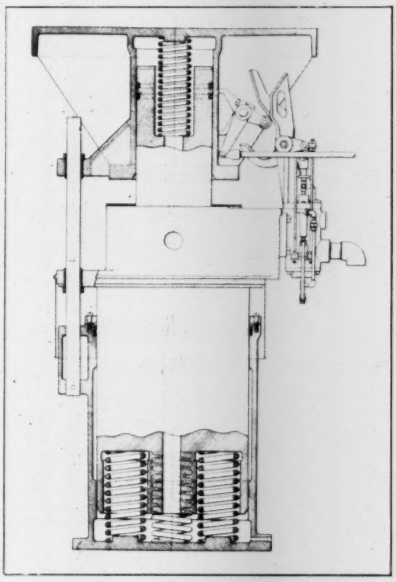

FIG. 2 SECTION OF JARRING MACHINE

20 As already stated, unlimited power may be expended in jarring sand to any given density; and since there is a certain maximum density corresponding to any given drop, it is also quite evident that efficiency increases with the drop, and decreases with the dead weight handled over and above the weight of sand used. But a certain amount of dead weight is inseparable from the process, and for this reason a heavy machine may not be used to its best advantage on light work. Nevertheless, with air as a working fluid the benefit gained by expan-

sion on light work offsets to a great extent the loss from the greater proportion of dead weight carried, giving to the jarring machine which uses air expansively in its cylinder quite a wide range of capacities, under approximately uniform efficiency as far as the consumption of air per cubic foot of sand rammed is concerned.

21 But it is not only the air consumed in lifting the loaded table that may not be utilized to the best advantage. At the instant when the loaded table strikes its anvil, the sudden change in the velocity of the table, whatever that may be, measures the pressure of impact, and the ramming effect is measured by the square of that change in velocity, which is proportional to the energy absorbed, part of which is utilized in ramming sand. Therefore, the greater the change in velocity at the instant of impact the greater the ramming effect, and by the laws of impact, the heavier the anvil the better. Efficiency in a plain jarring machine naturally increases with the weight put into the anvil; but since the cost of the machine depends very largely upon the weight of cast iron or concrete used, the weight of the anvil is generally limited to that of the loaded table. When the anvil is bedded on rock, it becomes practically of infinite weight, and capable of developing the maximum ramming effect for any drop given to the table. A rock bottom does not, however, eliminate the destructive ground waves, and often facilitates their transmission to unusual distances. To mitigate the effect of these shocks, the practice has been to bed the anvil on a timber cribbing, after the manner employed for steam hammers.

EFFICIENCY DEPENDS UPON THE ANVIL

22 So cushioned, the anvil when struck by a loaded table of its own weight will suddenly acquire one-half of the velocity of the table at the instant of impact, after which both table and anvil will be brought to rest by the yielding resistance of the timber cribbing; they will then be returned by its elasticity to their normal position. The loaded table, in this case, loses at the instant of impact only one-half of the velocity it would lose by falling upon an anvil of infinite weight, as exemplified practically in an anvil founded on rock. As a result, the retardation of the table by the compression of the wooden cribbing is less intense and less effective in ramming sand, although this second change in velocity no doubt has some effect, especially in the earlier stages of the ramming process while the sand is comparatively soft. Nevertheless the initial change in velocity, between a loaded

table and a floating anvil of equal weight, is only half as great as the change in velocity of a loaded table falling the same distance upon an anvil of infinite weight; and the ramming effect in the first instance, being measured by the square of the change in velocity, is only one-quarter as much as in the second case, where the whole energy in the falling mass is immediately absorbed.

23 An anvil cushioned upon a wooden crib may be considered as a floating anvil, in which the supporting medium is very dense and highly resistant, but in which also the resistance to compression is trifling compared to that of rock. The stiffness of such an elastic bedding for an anvil might be estimated from the anvil movement, of which no data are at present available; but, however effective it may be in the initial stages of the jarring process, it can have but little, if any, effect upon the final stages after the sand has been rammed to a density in excess of that corresponding to such elastic resistance. It may be said, without hesitation, therefore, that anvils cushioned upon wooden cribbing are much less effective than anvils founded in rock, and that such anvils, equal in weight to the loaded table, have in the final stages of the jarring process a comparative efficiency of only 25 per cent.

24 In considering the mechanical efficiency of a jarring machine it is therefore a matter of some importance to provide an anvil of maximum efficiency for any given weight. As a matter of course, the heavier the anvil in any case the better, and the unit standard for all anvils may be one of infinite weight, comparable to a foundation on rock. Such an anvil stands for the highest attainable efficiency, but it is not a practicable construction on account of the destructive ground shocks, which the shockless machine eliminates. We shall presently see how the anvil in this machine compares in efficiency with the usual type of anvil mounted on a wooden crib.

GENERAL DESCRIPTION OF THE SHOCKLESS JARRING MACHINE

25 The shockless jarring machine consists, in its usual form, of a jarring table mounted upon an upstanding plunger forming the anvil, which in turn is mounted in a cylinder base and supported upon long helical steel springs. Compressed air, as the working fluid, is admitted through an automatic valve, under hand control, attached to the plunger or anvil base, and passes first into the jarring cylinder to raise the loaded table. At some predetermined point in the table movement, the air is automatically cut off from the cylinder, and

while the valve is reversing, the confined air will expand and lift the table further from its anvil, provided its initial pressure exceeds the balancing pressure due to the weight carried. Then, when the operating valve completes its reverse movement the air from the jarring cylinder may be exhausted into the atmosphere, but preferably it passes from the jarring cylinder to the anvil cylinder beneath, and the table drops by gravity against the reduced pressure in its cylinder. At the same time, the plunger base or anvil is relieved of a considerable part of the load carried by its supporting springs, which immediately expand, giving the anvil an upward velocity to meet the falling table. When air is expanded from the jarring cylinder into the anvil cylinder this upward velocity of the anvil is augmented and the falling velocity of the table is somewhat retarded, but in any case the momentum of the rising anvil is substantially equal to that of the falling table at the instant of impact. As a result, both table and anvil come to rest with great jarring or ramming effect upon the sand, but without shock or jar upon the foundation or any surrounding material.

26 When the air from the jarring cylinder is discharged at once into the atmosphere the momentum of the falling table may somewhat exceed that of the rising anvil at the instant of impact; but when this air is expanded into the anvil cylinder it compensates more or less for the loss of spring pressure as the anvil rises, and in this case the momentum of the rising anvil may exceed that of the falling table at the instant of impact. The difference, however, need not be very pronounced, and simply results in a slight initial velocity of the table and anvil at the beginning of the next stroke.

27 The advantage of the second expansion is two-fold: it utilizes the potential energy of the compressed air in augmenting the momentum of the anvil, and at the same time it checks the acceleration of the table due to gravity and holds it in contact with the load upon it while falling. Otherwise a poorly fitted pattern board or flask may tend to spring away from its support while falling and cause lost motion, productive of a bad mold. For the same purpose, when the air is discharged directly from the jarring cylinder, a long compression spring between the jarring cylinder and its plunger may be introduced with good effect. In several instances such springs as shown in Fig. 2 have been made to carry half the weight of the table with 8-in. compression. They assist in lifting the loaded table, and retard its acceleration in falling; and by their use the lifting capacity of jarring tables may be considerably augmented. Their chief pur-

pose, however, is to retard the falling table and hold the pattern flask and sand firmly against it in readiness for the coming blow. With such a spring, the action of the table is, of course, somewhat slower in falling and more stroke is required to produce a given velocity of impact. On the other hand, the table rises faster and runs further to produce a given blow, and the increased stroke reduces the percentage of clearance space to plunger displacement. The spring in this position has, therefore, some beneficial effect upon the consumption of power, while serving a much better purpose, in the production of

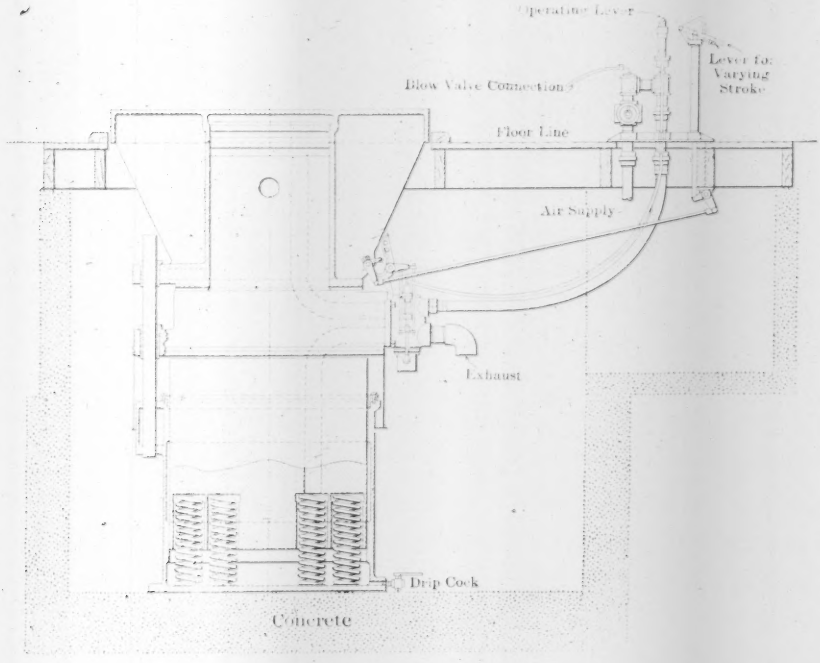

FIG 3. SECTION OF MACHINE TO HANDLE MOLDS WEIGHING 25 TONS

good molds, and although not so important when the air from the jarring cylinder passes through the anvil cylinder, it may be of some value in that case also.

28 The valve mechanism, and the means by which it is controlled, do not particularly concern the present discussion. It will suffice to say that the machine is started and stopped by an operating lever which controls the admission of air to the automatic mechanism. So long as this lever is held down, the machine will run automatic-

ally, and when released, the machine will stop. A latched lever is arranged to adjust the stroke, which can be varied while the machine is running. A safety stop is also provided to limit the table movement through the action of the main valve attached to the plunger base. When pressure is admitted to the jarring cylinder, the anvil cylinder opens to exhaust. When in action, it descends while the table is rising and then rises to meet the falling table.

29 Fig. 1 represents the design of a machine now being built for a large foundry to handle half-molds weighing 25 tons. The table is a steel casting 8 ft. x 12 ft., with lifting cylinder 3 ft. in diameter, and the plunger base forming the anvil is a solid iron casting weighing 65,000 lb. This is carried upon 22 steel springs, designed to compress 8 in. under the maximum load and to develop a working stress of only 60,000 lb. per sq. in., which is very much less than the usual working stresses on railway car springs and quite within safe limits. The total weight of the machine complete will probably be in excess of 90,000 lb. This is carried in a concrete pit designed simply to protect the machine and to support the static load on the floor of the pit.

30 The earthquake from a loaded table weighing 65,000 lb., dropping two to three inches upon an anvil bedded in the ground, can readily be imagined. Not only would it undo the work done by the machine, but a large area of valuable floor space in its vicinity would become useless and office buildings at a considerable distance might vibrate in sympathy. In this instance a comparatively small jarring machine of a well-known type, with anvil mounted on wooden cribbing had caused more or less annoyance to the occupants of office buildings in the neighborhood, and the machine above described was designed to avoid any further trouble of the same character.

31 It has been shown that a floating anvil which does not rise to meet the falling table, when equal in weight to the latter, has only one-quarter the efficiency of an anvil founded on rock. The efficiency of such an anvil, when mounted and actuated so as to acquire a momentum equal to that of the falling table at the instant of impact, remains to be determined. Obviously, the anvil will meet the table midway when the latter has fallen half the distance by which they were separated. In terms of the velocity of the table falling the whole distance, its velocity at this point will be $1/\frac{1}{2}$, and the velocity of the anvil will be the same. But in this case the velocity of the table is entirely destroyed, while in the previous assumption the change in velocity at the instant of impact was only half the final

velocity. The relative changes in velocity are, therefore, as 1 ½ to ½, and the ramming effects in the two cases will be to each other as the square of 1 ½ to the square of ½, or as ½ to ¼. Under the assumed conditions the anvil in the shockless machine is therefore twice as efficient as the same anvil cushioned on a wooden crib.

32 It might be shown still further that the vibratory action which develops equal momentum between the table and anvil is more efficient mechanically than any other action which develops unequal momentum. With compressed air as a working fluid, however, it pays to utilize its potential energy in the anvil cylinder rather than throw it away, and a decided gain in effect is realized in this way.

33 It may also be pointed out, that when the sand is soft the change in the velocity of the table at the instant of impact is greater than it is when rammed, because in the first condition the table movement is arrested before that of the sand, while in the latter condition, both stop together.

34 The loss of power in cylinder clearances is well known, and the obvious remedy of a short passage from the valve to the cylinder has led to the use of internal valves of more or less ingenuity and efficiency; but experience in machine design clearly points to the futility of attempting to embody all advantages or completely eliminate all disadvantages in any construction. The best machine for any purpose is the best compromise that can be made between conflicting advantages. Rather than save air by the use of a valve which is comparatively inaccessible, is it not better to sacrifice a little air for the sake of good construction, and accessibility to all working parts? At the same time, it may be said that the air consumed in the clearance passage to the jarring cylinder is not wholly wasted. It adds materially to the work done by expansion in the jarring cylinder, and again, when discharged into the anvil cylinder it adds to the momentum of the anvil. In addition to the consumption of air for any given stroke, it must not be forgotten that the blow struck in the shockless machine is twice as effective as the blow for the same expenditure of power in the usual type of jarring machine.

SPECIAL ADAPTABILITY IN HIGH BUILDINGS

35 Attention should also be called to the possibility of installing a machine of the shockless type on the upper floors of high buildings, where many foundries are now being located. The action of the machine is entirely free from jar except where it is wanted, on the

work produced, and the pulsating variation in floor load while running is no greater than is usually experienced in the operation of power squeezers. A number of these machines are now under construction for installation on upper floors, and in this connection it may be of interest to note that the original experimental machine, shown in Fig. 1, was set up on floor beams over a pit and operated without any vibration appreciable to a man standing on the beams while ramming up a half mold weighing about 1000 lb. In this case the weight of the machine was about 6000 lb., and a stroke of 4 in. was employed. As the movement of the anvil was about 1 in., it met the table when it had fallen about 3 in. The variation in the load in the floor beams was about 10 per cent of the static load carried, or between 600 and 700 lb. This variation, however, was gradual, the anvil rising and falling with the movement of the table. When impact occurred, the load on the floor beams simply ceased to decrease, and began again to increase without transmitting to the floor beams any part of the shock of impact, which was confined exclusively to the jarring table and its plunger base.

DISCUSSION ON LUBRICATION

LUBRICATION AND LUBRICANTS

By Dr. C. F. Mabery,[1] Published in The Journal for February 1910

ABSTRACT OF PAPER

A brief outline is presented of the history of lubrication, with allusion to the sources of the various lubricants. Petroleum products are briefly mentioned as to their composition and properties as lubricants. Graphite is referred to as possessing the most desirable qualities as an unctuous and extremely durable lubricant.

The use of the Carpenter frictional testing machine is explained, in testing light and heavy oils alone and with deflocculated graphite. Results are also presented on the efficiency of deflocculated graphite with water, kerosene and fuel oils as vehicles.

It appears that the coefficient of friction increases with increasing viscosity of lubricant, in close quantitative relations. The addition of 0.35% deflocculated graphite diminishes the friction of every oil and increases its efficiency and durability. Suspended in water, graphite maintains with no variation an extremely low coefficient of friction as do also kerosene and fuel oils.

Every lubricating oil has a well defined limit of load within which it can maintain a continuous film under definite conditions of oil supply and speed.

LABORATORY TESTS OF LUBRICANTS

Discussion by Dr. P. H. Conradson,[2] Franklin, Pa.

Non-Member

To make complete tests of lubricants—oils and greases—requires a great deal of expert knowledge and experience to enable the engineer to interpret correctly the results obtained. This point will be clearer perhaps, if one considers the various classes of machinery to be lubricated under all conditions of service, weather changes, and high and low temperatures.

2 The method of applying the oil to the parts to be lubricated, the condition of the bearing surfaces, composition of the journal and bearings, etc., play an important part int he proper interpretation of

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lubricating oil analysis. An oil that would give satisfaction when applied direct to the journal by means of soaked waste, might fail altogether if the wick method of feeding the oil were used; furthermore, a sight-feed cup with an orifice wide enough might give satisfaction, while a gravity feed through a long pipe of small bore might give very unsatisfactory results.

3 In making a complete investigation of the real or comparative value of a lubricating oil with another oil, we have then to consider the kind of machine to be lubricated, the service requirements and conditions, and the methods of applying the lubricant to the machine, and make our chemical or physical laboratory tests accordingly.

4 Generally speaking, the chemical tests, as made, are very inadequate, as are also the physical tests, especially frictional tests on oil-testing machines, unless the machines are constructed in such a way that the actual conditions can be approximately reproduced. For instance, in testing a spindle oil, the testing machine should be run practically at the same load and speed as the spindles are in actual service. In testing railway car and coach oils, the machine should have approximately the same size journal and bearing as would be found in actual railway car journals. The same is also true as regards speed, load, and the application of the lubricant.

5 While many valuable conclusions may be drawn from properly conducted laboratory tests, those of the greatest practical value come from a close observation of the lubricant in actual service, and we can base our laboratory investigations on the results obtained, especially in comparing different oils intended for the same work. To bring out the point more clearly, let us consider an air compressor, such as is used on street cars and on electric locomotives. As is well known, these compressors are not water-cooled or even air-cooled. It is not difficult to get an oil that will lubricate the compressor cylinder, but it is difficult to find an oil that will not carbonize at the high temperature, often 450 to 460 deg. fahr. in the street-car compressor, and 550 to 560 deg. fahr. in the electric locomotive air compressor.

6 The difficulty lies in the fact that as the compressed air passses through the outlet ports and check valves in the compressor heads, there is a very rapid increase in the temperature. The small amount of oil that goes with the compressed air, if not of the best or suitable quality will then begin to carbonize and cause trouble by forming heavy carbonaceous deposits on the check valves. Now, an ordinary oil-testing machine cannot bring out the essential requirements of a suitable oil for such service, and to make a proper laboratory investi-

gation and test of an air-compressor oil, it would be necessary to have an air compressor and test the oil as near as possible under actual service conditions.

7 Again, we may consider a steam turbine and suitable oils for its lubrication. Ordinary laboratory tests, both chemical and physical, such as are generally used in this country, do not bring out the essential qualities of a suitable turbine oil, because the service requirements and conditions are so entirely different from the general run of machinery, that special tests must be made. Therefore, from a practical point of view, to develop the essential qualities of turbine oils, it is necessary that the service conditions and requirements should be studied first, laboratory tests then being made in accordance therewith as far as practicable.

8 To illustrate, we may consider a steam turbine of the Curtis type, where the oil is forced under high pressure to the step bearing, and then returned to the oil tank. At first this might appear to be a very small matter, but in actual practice and experience it is not, for the following reasons: In the first place, leakage of steam occurs in most of these steam turbines as now constructed; this steam condenses, becomes mixed or churned in with the oil, and if the oil is not of the proper kind, it becomes emulsified. The emulsified oil gradually becomes thicker, and as the same circulating system is used for the rest of the machine, the emulsified oil often causes considerable trouble. Then again, we often find that the amount of oil used in the oil circulating system is entirely too small in quantity, so the oil has to pass through the turbine many times during the hour, in the twenty-four hours, and from week to week. This imposes a severe service requirement on the oil, which gradually becomes polymerized and oxidized, developing petroleum acids. If sulphur compounds are present to any extent, they become gradually oxidized and besides causing corrosion may cause a great deal of trouble from formation of asphaltic and tarry matter, which would clog the filters and orifices through which the oil has to pass. From a practical point of view, therefore, the laboratory tests of turbine oils should be considered along these lines. The same may be said of all lubricating oils intended for use in oil circulating systems, which are now so largely used in stationary power plants, shops and mills, war vessels, steam ships, etc.

9 It is a generally accepted idea that if the oil is adapted to the load and speed, the lower the viscosity, the better lubricant it will be. This, to my mind, holds good only where the service conditions

are uniform, and where the method of applying the oil to the bearing and journal is a positive one, such as in gravity or pressure pump systems. Where the climatic changes are great, as on railroads, this will not hold good.

10 The load and speed of the railroad trains are the same during the summer and winter, and as is well known, the practice in this country is to convey the oil to the journals by means of oil-soaked waste. A satisfactory thin winter oil with a low cold test and low viscosity, containing sufficient lubricating capacity to keep the bearing and journal apart, would not be suitable during the hot season, not because it has not the adequate sustaining power, but because of the method of applying it to the surfaces to be lubricated, making it necessary to use a much thicker oil than is theoretically required.

11 Therefore to make laboratory tests of the relative lubricating values of oils considered from a practical standpoint and to draw correct conclusions from the results obtained, we must consider the kind of machine or machines to be lubricated; the speed and the load; the composition of the bearing metal; whether the journals are iron or hard steel; the method of applying the lubricant, either with wick feed, soaked waste, sight-feed cups, flooded bearing or continuous oil-circulating system; the actual service requirements and the climatic conditions. We must make complete chemical and physical tests as near as possible in accordance with these conditions. I might with propriety state that one oil can not be considered a better lubricant than another oil unless the service conditions and requirements are specified and fully understood and the laboratory tests made in accordance therewith.

12 On the diagrams presented by Professor Mabery the first point of great interest seen is that his tests are run say, for two hours, to get all conditions uniform with a constant and given feed of oil; then the oil supply is shut off without stopping the machine, the tests being continued until the friction and temperature begin to rise rapidly, the time of the break being noted. The length of time from the shutting off of the oil supply to the break would, according to Professor Mabery's theory indicate the comparative lubricating value or endurance of the two oils, other things being equal; that is, if we take two oils intended for the same machinery and same service requirements and conditions, the oil that runs longer on the test machine after the supply is shut off, would be considered the better oil. I think that investigations can be made to great advantage in estimating the comparative lubricating value of two or more oils, by varying

the load and speed to a greater extent than Professor Mabery has done, as well as keeping the temperature of bearing and journal constant during the tests.

13 The second point of interest in Professor Mabery's results is the introduction of a very small amount of pure colloidal graphite in the oil to be tested. Especially interesting are the curves obtained from the mixture of water and graphite.

14 Professor Mabery called attention to the fact that crude oils from different sources as well as different treatment in the refining, produce lubricating oils of great variety, not only as regards the chemical composition and physical qualities, but also in lubricating value. Table 1 is of interest in this connection. Nos. 1, 2, 3, 4 and 5 are the same oil fractionated by means of "Florida Earth." While the flash and fire tests and the gravity remain practically the same, with changes in color and some also in viscosity, the congealing point or cold test has changed materially from the original No. 1, during the successive stages. From a technical and practical standpoint it would indeed be of great value, if Professor Mabery would make frictional machine tests (endurance tests) of these oils, to determine their relative lubricating values.

TABLE 1 LUBRICATING OILS

	1	2	3	4	5	6	7
Flash Point, °F.	365	370	370	370	370	410	415
Burning-Point, °F.	440	440	440	440	440	470	480
Gravity, 60°F.	30	30.7	30.5	30.3	30.2	23.0	22.8
Color, °Beaume.	Very dark	Yellow-ish	Orange	Red	Deep red	Red	Dark red
Cold Test, °F.	Zero	+18	+25	+25	+25	+5	+20
Viscosity (P.R.R.) pipette							
@ 125° F.	98	92	94	94	93	85	162
100°	160	149	148	153	151	143	309
90°	205	184	192	195	195	188	410
80°	271	238	249	254	254	253	Drops

15 Nos. 6 and 7 are the same oil before and after being in continuous service in an oil-circulating system about ten months. Here again the flash, fire and gravity tests remain practically the same. The color, as would be expected, has grown somewhat darker, but the congealing point and body or viscosity have greatly changed. The fresh oil had sufficient body or viscosity and gave satisfaction in service at the start, but its durability (endurance) in actual service, as seen from

the rapidly increasing sluggishness or viscosity, is seriously questioned Would Professor Mabery's endurance frictional tests, such as he conducted, have brought out the lack of stability of this oil? This is a practical question that practical men want to know, and to make

TABLE 2 TESTS ON GALENA SUMMER OIL

Bearing metal, brass; journal, steel; bearing surface: length 3.9 in., diameter 3.75 in., width 1.9 in., area 7.4 in.

Number of Test	1	2	3	4	5	6	7	8	9
Pressure on Journal, lbs. total.	1000	2000	3000	3500	4000	5000	5500	6000	6500
Pressure on Journal, lb. per sq. in.	135	270	405	473	540	675	743	810	878
Method of lubrication, flooding bearing.									
Minimum Coefficient of Friction.	020	019	0163	016	0156	0158	0153	0152	0151
Maximum Temperature of Journal, °F.	109	114	115	118	122	134	136	140	144
Temperature of Room, °F.	68	68	68	69	68	70	71	71	71
Elevation Temperature Journal above room	41	46	47	49	54	64	65	69	73
R. p. m. of Journal.	215	220	220	220	223	220	220	220	185
Feet traveled by rubbing surface per. min.	211	246	216	216	219	216	216	216	182

No. of Test	Time	R.p.m.	Temp. Journal °F.	Total Friction Lbs.	Coefficient Friction	No. of Test	Time	R.p.m.	Temp. Journal °F.	Total Friction Lbs.	Coefficient Friction
1	8.40	210	106	20	0.020	6	10.50	220	133	79.1	0.0158
	45	218	107	20	0.020		55	220	134	79.1	0.0158
	50	216	109	20	0.020		11.00	220	134	79.1	0.0158
2	9.00	220	112	38	0.019	7	11.10	220	135	84.0	0.0153
	05	222	114	38	0.019		15	219	136	84.0	0.0153
	10	218	114	38	0.019		20	221	136	84.0	0.0153
3	9.20	220	114	49	0.0163	8	11.30	220	138	92.0	0.0153
	25	220	114	49	0.0163		35	218	140	92.0	0.0153
	30	220	117	49	0.0163		40	221	140	92.0	0.0153
4	9.40	220	117	56	0.016	9	11.50	214	142	104.0	0.0160
	45	220	117	56	0.016		55	180	144	104.0	0.0160
	50	220	118	56	0.016		58	160	145	104.0	0.0160
5	10.00	216	120	63	0.0157	10*					
	05	218	121	63	0.0157						
	10	234	122	63	0.0157						

*Journal stopped at 11.58; pressure 6500 lb.

laboratory tests of real, practical value, problems of this nature must be satisfactorily worked out and answered.

16 In Professor Mabery's endurance tests on the Carpenter machine, only 150 lb. pressure per sq. in. was used on the test journal.

Table 2 gives results of tests of a Galena railway summer oil on the Carpenter oil-testing machine at Cornell University, with increasing loads on the journal from 135 lb. to 878 lb. per sq. in.; also a sort of endurance test to determine the load capacity of the oil.

17 From the results obtained, some interesting curves as to friction and temperatures could be plotted. The main point, however, is to show that with the Galena oil, after the pressure on the journal rises above 400 lb., and up to the maximum load used in the test, the coefficient of friction remains practically stationary, and would give a nearly horizontal curve From a practical standpoint this information is of great value.

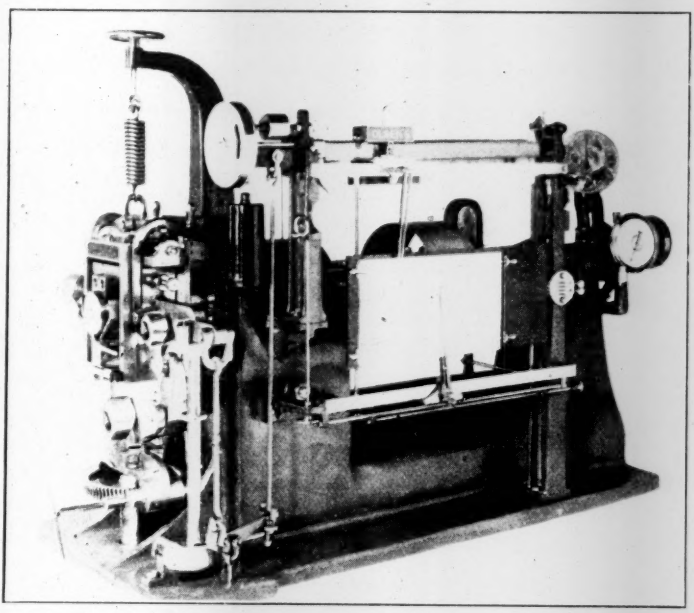

FIG. 1 OIL TESTING MACHINE, 20,000 LB. CAPACITY FOR BEARINGS UP TO 5 IN. BY 9 IN.

18 In connection with the foregoing a few comparative frictional oil tests made on the Galena-Signal Oil Company's oil testing machine will be of interest, because it is the largest and most complete oil-testing machine ever built. Full size M. C. B. car journal boxes and bearings can be inserted. By means of a system of levers, with screw and spring balance, varying loads up to 20,000 lb. can be applied on the test journal, while the machine is running in either direction at

any desired speed up to its maximum. The weight of the load is indicated on a dial; and the friction on the periphery of the journal, in pounds, is recorded on the scale beam in front of the machine. There is also a temperature indicator, a revolution counter and tacho-

TABLE 3 CONSTANT-TEMPERATURE TESTS

COMPARATIVE FRICTIONAL TESTS BETWEEN PURE RAPE-SEED OIL AND WINTER GALENA RAILROAD CAR OIL (ZERO COLD TEST OIL)

Steel journal, size 5 in. by 9 in. bearing, genuine babbitt; total load on bearing, 10,000 lb.; projected area 15.5 sq. in.; area of contact = 16.40 sq. in.

Pressure per sq. in. projected area.....645 lb.
Manner of lubrication.....oil bath
Average friction of four tests for each temperature

RAPE-SEED OIL			WINTER GALENA CAR OIL		
Temperature Deg. F.	Total Friction Lbs.	Coefficient of Friction	Temperature Deg. F.	Total Friction Lbs.	Coefficient of Friction
		300 r.p.m.=392.5 ft. surface speed			
50	21.06	0.00211	50	20.37	0.00205
70	15.375	0.00154	70	14.54	0.00146
90	13.875	0.00139	90	12.75	0.00128
		600 r.p.m.=785 ft. surface speed			
56	26.250	0.00263	60	19.94	0.00199
70	25.375	0.00254	70	18.31	0.00183
90	19.75	0.00198	90	15.31	0.00153

Viscosity P. R. R.		
@ 125°F.	80 Units	72 Units
100°	125 "	104 "
90°	141 "	125 "
80°	186 "	160 "
Cold Test	+15°F.	—5°F.

By constant temperature is meant that the oil-bath and bearing is kept at uniform constant temperature during the whole time of test, which lasts not less than one hour after the desired constant temperature of oil and bearing is reached.

meter, and an autographic arrangement showing the friction corresponding to the number of turns of the machine. The test journals and bearings are provided with a device for passing through either water or steam during the tests and there are arrangements for applying the lubricant by any desired method during the tests.

19 In Table 3 we find, first, three series of tests at constant but different temperatures; second, two series of great difference in speed (300 and 600 r. p. m.), all other things being the same; third, compara-

TABLE 4 CONSTANT-TEMPERATURE TESTS

COMPARATIVE FRICTIONAL TESTS BETWEEN WINTER AND SUMMER GALENA RAILROAD CAR OIL

Steel journal 5 in. by 9 in.; bronze bearing; 7800 lb. total load on bearing; 300 lb. pressure per sq. in.; 27.7 sq. in. area of contact; 363 r. p. m. 475 ft. surface speed. Manner of lubrication: oil bath. Average friction of four tests for each temperature.

	Galena Winter Car Oil	Galena Summer Car Oil
Temperature 65°F.		
Friction, right, lbs.	18.50 18.50	41.00 41.00
Friction, left, lbs.	18.50 18.50	41.50 41.50
Friction, average, lbs.	18.50	41.25
Coefficient of friction	0.00237	0.00529
Mean resistance per sq. in. of surface	0.665	1.489
Temperature 80°F.		
Friction, right, lbs.	15.50 15.00	29.50 28.50
Friction, left, lbs.	15.00 15.00	30.25 29.50
Friction, average, lbs.	15.125	29.44
Coefficient of friction	0.00196	0.00377
Mean resistance per sq. in. of surface, lbs.	0.546	1.063
Temperature 100°F.		
Friction, right, lbs.	11.25 11.00	20.00 20.00
Friction, left, lbs.	10.00 10.00	20.00 20.00
Friction, average, lbs.	10.563	20.063
Coefficient of friction	0.00135	0.00257
Mean resistance per sq. in. of surface, lbs.	0.382	0.724
Flashing Point, deg. F.	315°F.	395°F.
Burning Point, deg. F.	370°F.	455°F.
Gravity at 60 deg. F.	27.3°Beaumé	24.3°B.
Cold Test	+2°F.	+36°F.
Viscosity (P. R. R.)		
Pipette, 125°F.	83 Units	189 Units
100°	124 "	343 "
90°	156 "	473 "
80°	201 "	—

The mean resistance per square inch of surface is obtained by dividing the average total friction by the number of square inches (27.7) area of contact. See Note to Table 3.

tive tests of a purely vegetable oil (rape-seed) with a compounded petroleum oil (Galena lead-oxide process) with viscosities not far apart, as measured with the pipette viscosimeter; fourth, the difference in

friction of the two oils at the slower speed (300 r. p. m.) is very small, while the difference at the greater speed (600 r. p. m.) is considerable, the Galena oil having a much lower friction. This is very instructive when we consider the nature of these two oils, as well as the great load and speed.

20 Table 4 gives an interesting comparison between a winter and summer Galena car oil. Both have sufficient body to carry the heaviest load and speed in railroad service, but owing to the present method

TABLE 5 GALENA CAR, COACH AND ENGINE OILS

SHOWING WIDE RANGE OF VISCOSITY AND COLD TEST

Dudley viscosity pipette, 100 cu. cm. water at 60°F. (15.5°C.) 32 sec.
Redwood viscosimeter, 50 cu. cm. rape-seed oil 60°F. (15.5°C.) 535 sec.

Viscosity taken at 100°F., 37.7°Cel.

	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
Dudley	104	125	140	162	177	195	220	320	276	300	252	348	375	401	426
Redwood	151	183	204	240	260	283	335	378	392	444	480	510	556	595	650

Viscosity taken with Dr. Dudley viscosimeter–100 cu. cm. water at 60°F. (15.5°C.) 32 sec.
Instrument kept in air bath at same temperature as the oils.

Time in Seconds

125°F. (51.6°C.)	67	71	86	109	125	138	150	171	197
100° (37.7°C.)	96	99	132	185	214	247	277	318	370
90° (32.2°C.)	115	124	170	243	291	321	362	422	509
80° (26.6°C.)	141	156	216	318	380	439	501		

	Flash Point Open Cup	Burning Point	Gravity at 60°F. Beaumé	Cold Test
SUMMER				
Car	350°–380°F.	425°–450°F.	26.1 to 27.7°	+20° to +40°F.
Engine	350°–380°F.	425°–450°F.	25.4 to 26.4°	+20° to +40°F.
Coach	350°–380°F.	425°–450°F.	24.3 to 24.6°	+20° to +40°F.
WINTER				
Car	210°–300°F.	260°–380°F.	27.4 to 29.0°	−5° to +10°F.
Engine	210°–300°F.	260°–380°F.	26.6 to 27.9°	−5° to +10°F.
Coach	210°–300°F.	260°–380°F.	25.5 to 26.4°	−5° to +10°F.

of conveying oil to the car journals, a thick and sluggish oil with unnecessarily high viscosity is or has to be used during the warm or hot weather, naturally increasing the total train journal resistance, which of course, means excessive coal consumption. Note the great difference in the frictional resistance between these oils. Practical railroaders should ponder a little more on these facts and utilize such knowledge.

21 In connection with preceding tables, Table 5, is of interest. It gives a comparison between two viscosimeters, and illustrates the wide variation in congealing points or cold tests and viscosity possessed by first-class railroad lubricating oils, suitable to all services and climatic conditions. A close study of Tables 5 and 6 will be of great assistance to the practical user of lubricants.

22 I have spoken of the importance of adequate chemical tests in connection with physical and frictional tests. The following tests are therefore useful. While in some cases it is not necessary to subject

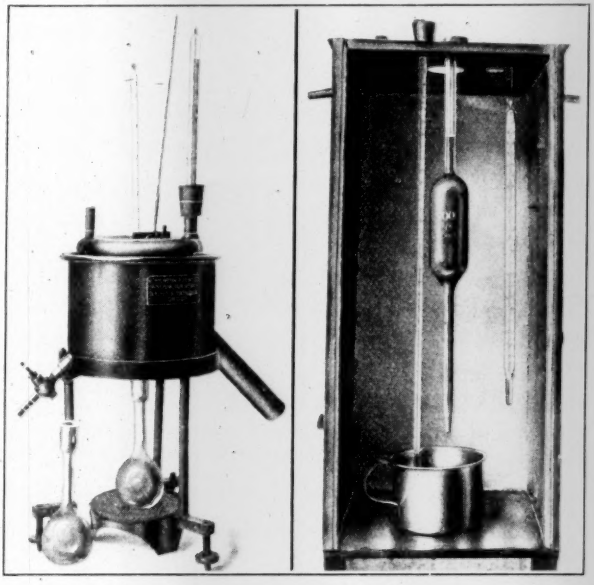

FIG. 2 REDWOOD VISCOSIMETER

FIG. 3 DUDLEY PIPETTE VISCOSIMETER

IN USING THE DUDLEY PIPETTE IT IS PLACED IN A CLOSED BOX WITH GLASS DOOR AND THE TEMPERATURE IN THE BOX IS KEPT CONSTANT

the oil to all these tests, they are of great importance in connection with special or unusual service requirements and conditions.

23 *Flash Point and Burning Point.* This test indicates the temperature at which the more volatile elements in the oil begin to vaporize to such an extent as to flash when a small flame is moved over and near the surface of the heated oil. The so-called open-cup method is generally used, and the heat is continued till the oil begins to burn when the test flame is applied. A flash and fire test that is too low

may be objectionable on account of danger from fire, besides causing too large a loss from evaporation under given service conditions. In connection with the viscosity, congealing point or cold test and gravity, the flash and fire test also enables the analyst to form an idea of the source of the petroleum.

24 *Gravity.* The test for gravity is a complementary test which enables the analyst in many cases to form an idea whether the oil is a Pennsylvania, Virginia or Western oil, the last generally having a much higher gravity.

25 *Color.* While not of much importance, other things being equal, an oil with a lighter color or pleasing appearance is oftentimes preferred to a very dark-colored oil.

26 *Odor* at times aids in detecting the kind and quality of fat oils in compounded oils.

27 *Purity* is freedom from water or matters in suspension, whether the oil is clear or turbid, etc.

28 *Gasoline Test.* This test indicates the presence of other foreign matter, or tar and asphaltic matter, if the oil gives a clear solution with 88 deg. gasoline before it is heated to burning point, but gives a precipitate with 88 deg. gasoline after this test. This indicates petroleum compounds which are readily acted upon by heat. Such an oil, in comparison with another oil, other things being equal, would not have the same lubricating value.

29 *Cloud Test* of a lubricating oil is sometimes of value in determining the amount of paraffines present and the behavior of the oil in chilling down to a temperature above congealing point.

30 *Cold Test--Congealing Point--Melting Point.* This test, together with comparative tests of the viscosity of the oil, is of much importance in connection with the service ability of the oil; and should be given careful consideration. The method used in determining the cold test and melting point, as well as getting at the comparative fluidity or sluggishness at temperatures lower than 70 deg. fahr., is not generally considered as it should be. To illustrate:

31 By the so-called P. R. R. method of cold test, valve or cylinder oil is frozen direct in an ice mixture and then stirred by the thermometer till it begins to flow when the bottle is inverted. By this method the oil may show a cold test of say 40 deg. fahr., and if no further observation is taken one has no idea of the fluidity at, say, 60 to 70 deg. fahr.; that is, the cold test would give 40 deg. fahr., but in reality the oil at 60 deg. fahr. would be so sluggish that it would hardly feed through the narrow bore feed pipe to the steam chest and

cylinders, and unless the analyst knows for a fact that the engineer has his pipe covered or warmed in some way, trouble may arise and the oil will be condemned, though it might be of the best quality.

32 Again, if the congealing point is taken by the Standard Oil Co.'s method, that is, the oil with the thermometer inserted in the bottle is put into a cooling box and gradually cooled till the oil just ceases to flow, then the bottle is inverted, or still better, the thermometer stem is lifted from time to time and note is made when the oil "hardly flows" from the stem. Without further observation however, this method,like the P. R. R. method, does not tell all. The rate of cooling or chilling—the time the oil remains in the chilling or cold-test box—plays an important part in proper interpretation of the comparative value of the oil in actual service.

33 *Viscosity.* This test is the bugbear of the oil tester—it may mean so much or so little. Certainly, in my work with the cold test and flash point, one might have a good idea of the quality and adaptability of the oil with the aid of a good viscosimeter.

34 As an adjunct to other tests, by careful study and knowledge of the service requirements, I have found the viscosity tests of the utmost value. In fact from my knowledge of all the analytical data, with the aid of the viscosimeter I can predict quite accurately the comparative friction of two oils under given conditions.

35 *Microscopic Tests.* In testing dark-colored oils, heavy machine oils, and cylinder oils, it is well to put a few drops on a slide and examine under the miscroscope. If carbonaceous matter is held in suspension, or if paraffine crystals are present at ordinary temperatures, by warming the oil a little the paraffine can be made to disappear, and other foreign matters held in suspension are brought out. Other things being equal, an oil that is free or practically so from black carbonaceous specks or flakes is certainly superior to an oil containing these in some quantity.

36 *Saponifiable Fats.* I will not discuss the method to determine these, but will merely point out that two cylinder oils, one containing thirty per cent of fat oil, and the other only 10 to 20 per cent of fat oil, other things being equal, while not of the same intrinsic commercial value, may have an equally good lubricating value. Again, a cylinder oil containing twenty-five to thirty per cent of good fat oil, might give excellent results in a steam engine at 100 to 150 lb. pressure per sq. in., the exhaust steam not being condensed or used over again in the boiler. Yet this oil might be very objectionable in connection with superheated steam and a surface condenser, the reason of course being obvious.

37 *Free Fatty Acids.* Other things being equal, the less free fatty acids are present the better.

38 *Petroleum Acids, Sulphuric Acids, Chemicals.* The presence of pertoleum acids, sulphuric acid, sulphonates and chemicals from imperfectly refined petroleum oils should always be carefully investigated, as the presence of these foreign materials in a lubricating oil, at least for certain services, might lead to serious trouble and complications. A first-class lubrication oil should be free from, or at least contain only traces of these impurities.

39 *Sulphur.* In general very little attention is paid to sulphur and organic sulphur compounds that may be present in lubricating oils. In the future, the sulphur in lubricating oils will have to be reckoned with when these oils are used for turbine service, or where the oil is used over and over again as in an oil-circulating system. Under these conditions the oil, due to the continuous exposure to heat, air, moisture and metal bearings, gradually becomes oxidized and polymerized, forming acid petroleum products, which change it both chemically and physically. The sulphur compounds present in the oil largely augment the corrosive or pitting action on the bearings and journals.

40 In the examination of lubricating oils for sulphur contents, it is important to make a distinction as to how they occur in the oil. I have often found that in making sulphur tests by burning a given amount of oil in lamps to take up the products of combustion in a carbonate of soda solution, it is necessary to consume all the oil in the test lamps, and to make a determination of the sulphur compounds left in the wick. In some poorly chemically refined lubricating oils, the sulphur compounds found in the wick oftentimes amount to from twenty to forty per cent of the total sulphur present.

41 *Maumene Test.* This is the sulphuric acid thermal test, and is of value in connection with tests of compounded lubricating oils.

42 *Evaporating Tests.* By exposing the oils in shallow flat-bottomed dishes in an air bath at 212 to 300 deg. fahr. for six hours, noting the percentage of loss and condition of residue and its behavior when mixed with 88 deg. gasoline, we obtain valuable information as to the amount of volatile matter at low temperatures.

43 *Heat Tests.* For certain service such as air compressors not water-cooled, turbines, etc., valuable data may be obtained by exposing the oils in shallow flat-bottomed dishes in a covered air bath through which air is blown for six hours, at temperatures of 425 to 450 deg. fahr.; studying the residue in the dish by dissolving the same in 88 deg. gasoline, noting whether the gasoline solution is clear or turbid, and the amount of precipitate, if any, on standing.

44 *Emulsifying Tests.* To determine the adaptability of an oil for lubrication in turbines of the Curtis type (step-bearing) it is of the utmost importance to ascertain the behavior of the oil when coming in contact with steam through the step-bearings, whether it forms a thick creamy emulsion or separates readily from the steam and condensed water.

45 *Tabulation of Chemical Tests:*

Flash point
Burning point
Gravity
Color
Odor
Purity
Gasolene tests, before and after flash
Cloud test
Cold test
Viscosity
Microscopic test for carbonaceous matter in suspension
Saponifiable fats
Free fatty acids
Petroleum acids
Sulphuric acid
Chemicals from imperfect refining
Sulphur-lamp test and in wick
Maumene test
Iodine test
Evaporating tests, a given time at 200 to 300 deg. fahr. to study percentage of volatile, and behavior of residues in 88 deg. gasolene tests and acidity.
Heat test, in air bath blowing air over the oil at 425 deg. fahr. and 540 deg. fahr. Examination of residue.
Emulsifying tests, to determine adaptability of the oil, say in turbine service
Tar and coke-forming elements present before and after heat test
Oxidation or gumming tests.
Superheated steam tests
Carbonizing tests in connection with air compressor (not water-cooled) automobile gas-engine lubrication
Capillarity or wick tests.

46 *Frictional tests.* To make frictional tests of oils and greases of practical value requires testing machines constructed for various loads, speeds, sizes of journal and bearings, and methods of applying the lubricant, comparable to those in actual service, as well as devices to keep journal and bearing at any desired constant temperature during the tests.

47 The constant-temperature tests are of importance not only for the purpose of standardizing the machine to get all conditions of bearing, journal and feed, properly regulated before the actual tests begin, but equally so for making comparative frictional tests of two oils of approximately the same viscosity. The two oils may show practically the same friction at a given temperature, but to keep the journal and bearing at this temperature, one may require a great deal more water or steam passing through the journal and bearing; again the friction may be practically the same at a temperature

of say 150 or 125 deg. fahr., but very different at 90 or at 70 deg. fahr. The constant-temperature frictional tests are therefore of great value in conducting comparative tests.

48 As a rule the reports of frictional tests are very incomplete; they should give all the constants and data taken, such as area of contact, projected area, total pressure on journal in pounds, pressure per square inch in pounds, total maximum, minimum and average friction in pounds, coefficient of friction, temperature of journal and bearing, number of revolutions and feet traveled by rubbing surface per minute, duration of tests, constant or freely increasing temperature, amount of lubricant and method of feed, besides complete analytical chemical data.

49 Where the service conditions are uniform or fairly constant, as in mills or power houses, the comparative viscosity and the congealing or fluidity points and friction, other things being equal, would determine the most economical or suitable oil or grease for the service. But in making comparative tests, chemical, physical and frictional, of lubricants under varying service conditions, expecially climatic conditions, it should be borne in mind that two oils showing considerable difference in viscosity, congealing or fluidity, and friction, may be equally good for the service requirements. Manufacturers should submit for comparative tests, samples intended to do service not for the whole year but for the different seasons.

50 In conclusion, what function should a lubricant perform? What are the necessary requisites or qualities that should be inherent in a first-class lubricating oil? These are trite questions, and will be answered briefly:

51 First, the function of a lubricant is to keep the rubbing surfaces (journal and bearing) apart to prevent undue abrasion, friction and heating.

52 Secondly, the necessary requisites or qualities that a first-class lubricating oil should possess in a high degree may be enumerated as follows: Necessary body to withstand the severest pressure in the service for which the oil is intended, so as to keep the rubbing surfaces apart, forming a continuous film between them, filling up the inequalities in the surfaces; the quality of spreading itself rapidly over the rubbing surfaces, with the requisite degree of adhesive power to remain between the rubbing surfaces without creating undue friction and heating; requisite mobility or fluidity at all seasons of the year, and in all climates from the coldest to the hottest, without impairment of the necessary intrinsic lubricating body for the required service;

durability; freedom from mineral and organic impurities, tarry and asphaltic matters; non-drying, non-gumming.

53 Yet no matter how excellent and suitable a material or machine may be, if it is not properly applied or used, the best and most economical results are not obtained. This has brought about the idea of the oil manufacturer's employing practical and experienced men educated in actual service to follow up and watch the proper application and economic use of the various lubricating oils and greases. These men have demonstrated, not only to their employers, but also to the consumer, the practical and economic value of their educational work.

54 The importance of this "following up" is far-reaching. It has gradually brought about a much more systematic and uniform method in lubrication; it has brought about greater economy in the consumption of lubricating oils, and at the same time demonstrated the possibility of better lubrication. In fact, in many instances the consumption of oil has been reduced from 50 to 100 per cent, without impairment of the best and most economic lubrication.

55 From these remarks, you will readily appreciate that to make laboratory tests of lubricants of real practical value, not only to the consumer but also to the manufacturer, involves considerable technical and practical knowledge and experience, besides full and complete laboratory equipment and experience; and the chemist or engineer who is called upon to give a qualified opinion as to the relative, comparative lubricating values of two oils or greases for a given service, considered from a practical and economical service standpoint, has indeed a difficult and oftentimes thankless task to perform.

FURTHER DISCUSSION ON LUBRICATION

WILLIAM M. DAVIS.[1] I have not had very much experience with oil-testing machines, preferring to determine the lubricating value of two oils by actual use on the engines or machinery, assuming, of course, that the oil which will keep the bearing cooler is the better lubricant. In my capacity as oil inspector for a corporation in the New England States operating a large number of mills, I found that in some of the mills a heavy engine oil, compounded with about 10 per cent of lard oil, was used on their engines, while the manufacturing department used an excellent grade of straight petroleum machine oil.

[1] Lubrication Engineer, 93 Broad Street, Boston.

2 It occurred to me that the machine oil, which cost about ten cents a gallon less than the engine oil, was quite good enough to use on the engines. One objection to the compounded oil, however, was that it caused gumming in the cups, and sometimes made it difficult to clean the generator and motor coils when gummed on them.

3 In one engine room a thermometer placed in the main bearing of the engine while using the engine oil showed a maximum temperature of 135 deg., the temperature of the room being 80 deg., a rise of temperature due to friction of 55 deg. Readings were taking hourly from the time the engine started in the morning until shutting down at 6 p.m.

4 A bearing under ordinary running conditions will reach a point where the temperature remains constant, the heat radiated equaling the heat generated, as long as the oil feed, the speed and the pressure or load are constant. From the results of the test, Curve A, Fig. 1,

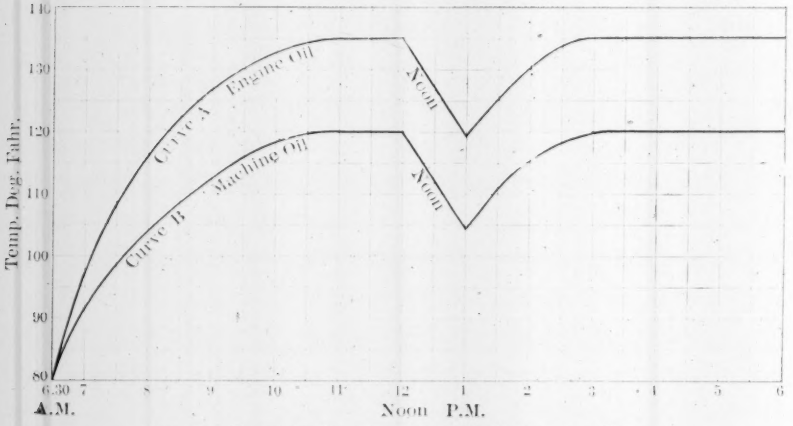

FIG. 1 CURVES PLOTTED FROM TESTS ON OILS

was plotted. As the engine was shut down for forty-five minutes at noon the bearings cooled off a few degrees, but reached the maximum temperature again in an hour or so and remained constant the rest of the day. The temperature of the room remained constant at about 80 deg.

5 The next day the test was repeated, using the machine oil. As will be seen by Curve B, the maximum temperature was only 120 deg., the temperature of the room remaining at 80 deg., showing a rise of temperature due to friction of only 40 deg. This test was repeated on several different engines at different times and the results

were practically the same in each case, proving conclusively that the machine oil was a better lubricant.

6 I do not attribute the higher temperature while using the engine oil wholly to the fact that it contained lard oil, but to the fact that it was considerably higher in viscosity than the machine oil. The viscosity of the engine oil (by Saybolt viscosimeter) was 240 seconds at 100 deg. fahr., and that of the machine oil 178 seconds at 100 deg. I prefer to make the tests in this way, for as Professor Mabery and Dr. Conradson have said, the results of tests on friction-testing machines may not agree with results obtained in actual service.

7 Another test under somewhat similar but more severe conditions showed that the straight petroleum oil kept the bearings cooler than the compounded oil. In this case, however, the oils were of the same viscosity and flash tests. The tests were made on the 22-in. main bearings of a large compound engine making about 120 r.p.m.—a very high speed for a shaft of this size. The engine was fitted with a continuous oiling system, the oil draining back into a chamber under the bearings, no cooling coils being provided to reduce the temperature of the oil. The temperature of the oils flowing on the bearings was 130 deg.

8 On August 25, while using the compounded oil, the maximum temperature of the bearing was 170 deg.—a rise of temperature due to friction of 67 deg., the temperature of the room being 103 deg. The next day while using the straight petroleum oil the maximum temperature was 148 deg., and the following day it was 138 deg., the temperature of the room being 88 deg.,—a rise of temperature due to friction of 54 deg. and 56 deg. respectively. These results showed that the petroleum oil kept the bearings cooler than the compounded oil.

HENRY SOUTHER. In the evolution of the automobile business I was called upon to approach the problem of lubricating the automobile engine. The needs are peculiar and absolutely different from those of the steam engine, and very different from those of the large gas engine. I went at the problem in the belief that as high temperatures were to be met, high flash points would be necessary. I very soon discovered that high flash points meant residue oils of high viscosity, and that these oils gummed in the cylinders to such an extent that although they lubricated the engine fairly well while running, the engine very soon refused to run because clogged with tarry matter.

2 A lighter oil was then tried in the cylinders, and a heavy oil in

the crank bearings. These worked well enough until the two oils began to mix. The oil resulting was better than the thick oil formerly used, but still unsatisfactory. I then proceeded to obtain a large variety of oils and try them out. The equipment of my laboratory included a four-cylinder engine running against a dynamo brake; and I had the use of several automobiles. I knew from experience that one of them was easy to lubricate, while another was exceptionally difficult. The engine of the latter was very sensitive, being of very high compression and very close in all its clearances.

3 I soon learned that oils that gave good practical results in an automobile engine possessed certain characteristics; not those obtained with an oil-testing machine, but obtainable with a viscosimeter, with flash and fire tests and with the Westphal balance (specific gravity). But even these tests did not seem to tell the whole story. I had read more or less about certain carbon-residue tests, but rather doubted the efficacy of any such test and the possibility of duplicating any results by destructive distillation of the oil. Nevertheless, I gave instructions to a laboratory assistant, and though he was skeptical at first, he reported in two or three weeks that he thought he could duplicate the results and prove it to me. I checked him by giving him unknown samples and duplicate samples, and he obtained results ranging from 0.1 per cent to 1.5 per cent total residue and checked closely.

4 Thus from these tests I learned what I still consider the desirable characteristics of an automobile-engine lubricant. This is a lubricant which will take care of the main shaft bearings, of the crank-pin bearings and of the piston, and one which when it passes up into the explosion chamber will disappear as much as possible; that is, will leave as little residue as possible. Whether the residue is carbonaceous or not I do not pretend to say; it looks like carbon in a coke-like condition. However, a good oil, as judged by these characteristics, will leave less residue than a bad oil.

5 I measured the viscosity both hot (210 deg. fahr.) and cold (70 deg fahr.). The viscosity when hot, measured by the Saybolt viscosimeter, should be between 40 and 50 seconds. Tested cold (70 deg. fahr.) it should not be over 300 seconds. This latter viscosity limit is placed to exclude the use of residue oils. No distillate oil can be made that will exceed 300 seconds viscosity at 70 deg. fahr., so that my specifications exclude absolutely a residue oil and accept a distillate oil.

6 As to the specific gravity there is some doubt. I am inclined to favor a high Beaumé test—between 28 and 31.

7 In the carbon-residue test I have set an upper limit of 0.5 per cent. In this test the personal equation is admittedly considerable, as is also the fact with many other tests of a practical nature. A number of assistants in my laboratory have been so trained that they can duplicate each other's results without trouble. I believe that other chemists or other testers can duplicate results by following the written instructions for this test, which is more than can be done with some chemical operations.

8 However, tests of the kind referred to in Professor Mabery's diagrams are absolutely valueless in connection with the lubrication of an automobile engine, in which the main crank bearings, the pins and the main journals are flooded with oil.

9 The popular method of lubrication, and I think perhaps the most successful, is the circulating system, which keeps the oil flowing to and from the bearings. The only objection to all systems is that the oil will work into the explosion chamber and leave a coke-like deposit or residue, which when heated to incandescence causes premature ignition. In that case smooth running is impossible. I think that a high compression engine is much more sensitive to this trouble than a low-compression engine. With the former type the temperatures seem to be so much higher that the so-called carbon deposit forms much harder and remains incandescent longer. That may not be true, but it seems to be true from my experience.

FRED R. LOW. Some months ago *Power* was asked how much oil was ordinarily used to lubricate the cylinders of steam engines. An appeal to our readers brought out numerous statements concerning individual practice, eighty cases of which, as bearing upon the practical side of this discussion, are here tabulated so as to compare the amounts and costs per horsepower-hour and per million square feet of cylinder surface passed over.

2 A wide variation in these results, affected as they are by the design of the engines, the nature and finish of the surfaces, the quality, pressure and temperature of the steam, the adaptability of the oil to the service, method of application, and above all, degree of care and intelligence exercised in its use, would naturally be looked for, and it is there. If we compare the amounts of oil used per million square feet of surface wiped over by the piston, we find one man using 0.07 of a pint and another 5.94 pints, or 85 times as much, to lubricate the same area.

3 The higher value here mentioned, 5.94 pints, is, however, ex-

ceptional, the next highest value being 2.59 pints, and in the eighty cases analyzed only four go above 2 pints. In the high case the steam was very wet, which indicates that there are advantages to be gained by dry-steaming boilers, steam-pipe coverings and separators, besides the saving of heat units and cylinder heads. The low value, 0.07, is also very far from the average, which is 0.875, only eight of the cases being below 0.25.

4 The monetary importance of this question of cylinder lubrication is indicated by the column of costs per horsepower-hour, the average of which is 0.0075 of a cent. On a 1000-h.p. plant running 3000 hours per year, this would amount to $225 per year, and on a continuously running plant to between two and three times as much. For the high value, with wet steam, it costs 0.07 of a cent per h.p.-hr., which would be $2100 per 1000 horsepower, and a 3000-hr. year; while in the lowest case, at 0.00047 of a cent per h.p.-hr. the cost would be $14.10, a difference which would more than pay the wages of the average engine attendent in this one item alone.

5 It goes without saying that a very small part of the oil ordinarily used would lubricate the surfaces if it could be minutely and specifically applied to the surfaces in moving contact. Water is an effective preventative of such application. Oil will not attach itself to a wetted surface any more than water will adhere to an oily one.

6 The practical men to whom I talk are divided as to the best means of application. Some want the oil atomized, arguing that if it is diffused throughout the steam it will be carried to all the surfaces. Others prefer to keep the oil together and let it trickle down the side of the pipe and be washed along over the surfaces by the flow of steam, arguing that it is the valve and cylinder surfaces and not the exhaust pipe that they want to lubricate. Some prefer to use a force pump with a multiplicity of feeds, leading the oil positively to the various points of application. Some go so far as to drill the seats of Corliss valves longitudinally with several openings to the under surface of the valve; others, admitting the positiveness and efficiency of such systems, claim that they can get their oil where they want it with a simple sight-feed cup in the steam main, with fewer joints to leak, passages to clog and bright work to polish. The idea is prevalent among them that there is apt to be less trouble from a moderately smooth cylinder, one which shows the tool marks, than a highly polished one, the minute rugosities of the less highly finished surface aiding in the retention of the oil. This may have a bearing upon the fact often mentioned, that cylinders that have been run successfully without oil, as

No.	Estimated Horse-power	Description of Engine	Cylinders	Stroke	R. P. M.	Steam Pressure	Sq. Ft. Rubbed over by Piston per Hour
1	1350	Corliss triple expansion	20-34-52	60	65		1,084,000
2	675	Cross compound	h.p. 26	48	81		265,800
3	675	Cross compound	l.p. 52	48	81		531,600
4	975	Cross comp. St. Louis Corliss	22-44	48	85	140	706,500
5	975	Corliss compound	24-44	60	65		694,500
6	650	Hamilton Corliss cross comp.	20-36	44	100	140	644,400
7	650	Hamilton Corliss cross comp.	20-36	44	100	140-100 sup.	644,400
8	575	Corliss compound	18-34	48	85		555,900
9	875	Corliss tandem compound	24-42	48	65	110	539,600
10	450	Russell cross compound	16-30	24	150		433,300
11	650	Cross comp. condensing	18-36	42	72		428,100
12	290	Cross compound	h.p. 18	36	86	120	146,000
13	290	Cross compound	l.p. 34	36	86	120	276,000
14	200	Harris standard cross comp.	h.p. 16	18	200	110	150,800
15	200	Harris standard cross compound	l.p. 28	18	200	110	263,900
16	400	Cross comp. St. Louis Corliss.	14-28	42	85		393,000
17	450	Corliss cross comp.	18-30	36	84		380,000
18	290	Phoenix Iron Works	14-24	18	210	130	376,100
19	200	Russell tandem compound	13-20½	20	210		368,200
20	225	Corliss cross compound	h.p. 16	36	84		126,800
21	225	Corliss cross compound	l.p. 30	36	84		237,500
22	225	Buckeye tandem comp.	12-21	21	200		363,100
23	450	Ingersoll cross compound	18-30	24	100		302,000
24	200	American Ball compound	10½-20½	12	285		277,600
25	180	Westinghouse automatic comp.	11-19	11	300		259,300
26	160	Cross comp. Meyer valves	11-18	14	165	115	175,400
27	160	Tandem comp. piston valve	10½-18	18	115	115	172,100
28	70	Ingersoll cross compound	8-12	12	150		94,300
29	45	Piston valve	30	48	95	150	357,800
30	450	Double eccentric Corliss	30	48	84	150	318,000
31	290	Corliss	24	48	102		308,100
32	290	Harris Corliss	24	48	100		302,000
33	290		24	42	93	120	245,500
34	200	Greene	20	44	100		230,200
35	160	Sioux Corliss	18	42	105		207,900
36	290	Harris Corliss	24	48	65		196,200
37	200	Corliss	20	36	102	150	192,200
38	130	New Brown	16	36	125	110	188,800
39	250	Atlas single valve automatic	22¾	30	105	110	183,500
40	160	Ames	18	16	230		173,700
41	160	Corliss	18	36	100		169,800
42	130	High speed automatic	16	16	250	115	167,800
43	130	High speed automatic	16	16	250	115	167,800
44	150	Piston valve engine	17½	16	220	80	161,500
45	130	Slide valve throttling governor	16	30	126	95	158,200
46	120	Single valve auto. Fitchburg	15½	14	275		156,200
47	160	Atlas four valve	18	15	220	125	155,900
48	130	Harris Corliss	16	36	100	65	150,900
49	130		16	38	92		146,600
50	100		14	14	280		143,200
51	240		22	23	106		140,500
52	160	Wright Corliss	18	42	70	85	138,800
53	130	Nordberg Corliss	16	36	90		135,800
54	100	Robt. Armstrong automatic	14	14	260	125	133,500
55	130	Corliss	16	42	75		132,100
56	100	Corliss	14	36	100		132,000
57	200	Slide valve	20	30	80	80	125,800
58	160	Greene	18	30	85		120,200
59	100	Variable speed, St. Louis Corliss	14	36	90	135-140	118,800
60	85	Atlas automatic	13	18	190	125	116,800
61	110	Ames	15	14	212		116,600
62	100	Automatic piston valve	14	14	223		114,500
63	130	Nordberg Corliss	16	32	85	110	113,800
64	70	Ideal	12	12	200	100	113,100
65	70	Buckeye	12	21	165		108,900
66	100	Corliss	14	36	80		105,800
67	70	Atlas single valve automatic	12	18	180	110 very wet	101,800
68	70	Corliss	12	36	90		101,800
69	70	Bates Corliss	12	36	88	100	99,500
70	70	St. Louis Corliss	12	36	85	135-140	96,350
71	70	New Brown	12	34	90	80	96,200
72	50		10	14	260	80	95,400
73	100	Atlas	14	20	130	80	95,400
74	50	McEwen	10	12	300		94,400
75	70	Fitchburg	12	30	100	90	94,400
76	70	Slide valve	12	14	210	150	92,500
77	40	Ball	9	12	300		84,750
78	50	Atlas	10	12	240		75,400
79	50	Center crank	10	15	170		66,800
80	85		13	17	105	100	60,900
81	50	Slide-valve	10	12	250	100	31,410

Averages

Rub-r by per r	Oil Used — Name or Description	Price per Gal. Cents	Total Amount Used per Hr., Pints	Cost per Hour Cents	Amt. Used per Estimated 1000 h. p. hr. Pints	Cost per Estimated 1000 h. p. Hr., Cents	Amt. per 1,000,000 sq. ft. Rubbed over per Hr., Pints
00	.....	28	0.69	2.42	0.511	1.79	0.636
00	No. 725 cyl. comp.	75	0.466	4.37	0.69	6.46	1.76
00	Capitol oil	35	0.666	2.91	0.99	4.33	1.25
00	No. 650 dark valve cyl.	50	0.104	0.65	0.107	0.67	0.147
00	.....	28	0.621	2.17	0.639	2.23	0.89
00	"600 W"	60	0.21	1.57	0.323	2.42	0.33
00	Oil of beeswax; 600 fire test Cyl. stock & acidless tallow	90	0.255	2.87	0.392	.....	0.40
00	.....	28	0.465	1.63	0.81	2.84	0.84
00	Heavy body comp.	40	0.2	1.00	0.229	1.14	0.37
00	"600 W"	60	0.25	1.87	0.555	4.16	0.58
00	Improved high pressure cyl.	75	0.111	.....	0.171	.....	0.26
00	Best grade cyl.	60	0.645	4.34	2.22	15.80	4.42
00	Best grade cyl.	60	0.323	2.42	1.11	8.32	1.17
00	Harris H. P. valve	60	0.184	1.38	0.92	6.90	1.22
00	Rarus	.....	0.131	.....	0.65	.....	0.50
00	No. 650 dark valve	50	0.09	0.56	0.225	1.41	0.23
00	"600 W"	60	0.417	3.13	0.927	6.95	1.10
00	.....	43	0.167	0.90	0.58	3.10	0.44
00	"600 W"	60	0.167	1.25	0.835	6.25	0.45
00	Best grade cyl.	60	0.645	4.34	2.86	21.45	5.10
00	Best grade cyl.	60	0.323	2.42	1.43	10.72	1.36
00	"600 W"	60	0.25	1.87	1.111	8.33	0.69
00	"600 W"	60	0.167	1.25	0.371	2.78	0.55
00	.....	.....	0.273	.....	13.65	.....	0.98
00	No. 725 cyl. comp.	75	0.083	0.78	0.461	4.32	0.32
00	Best high grade mineral	50	0.012	0.075	0.075	0.47	0.07
00	High grade mineral	50	0.037	0.23	0.231	1.44	0.21
00	"600 W"	60	0.083	0.62	0.185	8.89	0.88
00	"Cyl. oil No. 10"	25	0.417	1.30	0.927	2.90	1.16
00	"Cyl. oil No. 10"	25	0.5	1.56	1.111	3.47	1.57
00	.....	.....	0.091	.....	0.314	.....	0.29
00	.....	.....	0.091	.....	0.314	.....	0.30
00	.....	60	0.369	2.77	1.272	9.55	1.50
00	.....	.....	0.91	.....	0.455	.....	0.39
00	"600 W"	60	0.125	0.94	0.781	5.86	0.60
00	.....	.....	0.19	.....	0.655	.....	0.97
00	"Cyl. oil No. 10"	25	0.167	0.52	0.835	2.61	0.87
00	Harris high-press. valve	60	0.118	0.88	0.908	6.81	0.63
00	Buckeye cyl.	28	0.15	0.525	0.600	2.1	0.82
00	Capitol oil	35	0.22	0.96	1.375	6.00	1.27
00	.....	90	0.1	1.12	0.625	7.04	0.59
00	High grade mineral	50	0.015	0.094	0.115	0.72	0.089
00	High grade mineral	50	0.022	0.14	0.169	1.06	0.13
00	Light colored oil	.....	0.03	.....	0.2	.....	0.19
00	Capitol	35	0.1	0.44	0.769	3.36	0.63
00	.....	28	0.138	0.48	1.15	4.02	0.88
00	Rarus cyl.	.....	0.094	.....	0.587	.....	0.60
00	Harris std. grade	60	0.15	1.12	1.153	8.64	0.99
00	Compounded oil	43	0.15	0.81	1.153	6.20	1.02
00	Model cyl.	.....	0.045	.....	0.45	.....	0.31
00	"Eureka cyl."	.....	0.105	.....	0.437	.....	0.75
00	Capitol	35	0.125	0.55	0.781	3.41	0.90
00	Valvaline	75	0.2	1.87	1.538	14.38	1.47
00	"600 W"	60	0.15	1.12	1.5	11.22	1.12
00	.....	28	0.138	0.48	1.061	3.71	1.01
00	"600 W"	60	0.06	0.45	0.6	4.50	0.45
00	.....	.....	0.018	.....	0.09	.....	0.14
00	"600 W"	60	0.03	0.22	0.187	1.46	0.25
00	No. 650 dark valve cyl.	50	0.045	0.28	0.45	2.82	0.38
00	"600 W" vacuum cyl.	60	0.067	0.50	0.788	5.90	0.57
00	Capitol	35	0.11	0.48	1.0	4.37	0.94
00	High grade heavy	50	0.125	0.78	1.25	7.8	1.09
00	Dark heavy cyl with graph.	.....	0.115	.....	0.885	.....	1.01
00	Capitol cyl.	35	0.083	0.36	1.186	5.18	0.73
00	.....	.....	0.1	.....	1.428	.....	0.92
00	.....	.....	0.049	.....	0.49	.....	0.47
00	Buckeye cyl.	28	0.6	2.1	8.57	30.0	5.94
00	.....	.....	0.0835	.....	1.19	.....	0.82
00	Capitol cyl.	35	0.2	0.87	2.857	12.48	2.01
50	No. 650 dark valve cyl.	50	0.045	0.28	0.643	4.01	0.47
00	Franklin oil	75	0.1	0.94	1.428	13.38	1.04
00	Capitol cyl.	35	0.055	0.24	1.1	4.81	0.58
00	Capitol cyl.	35	0.2	0.87	2.0	8.74	2.10
00	W. P. Miller's cyl. comp.	75	0.04	0.37	0.8	7.50	0.42
00	.....	45	0.055	0.31	0.786	4.42	0.58
00	Flake graph. with eng. oil	.....	0.083	.....	1.185	.....	0.90
50	Pomo oil	.....	0.167	.....	4.175	.....	1.97
00	.....	60	0.045	0.34	0.9	6.75	0.60
00	.....	.....	0.083	.....	1.66	.....	1.24
00	"600 W"	60	0.125	0.94	1.47	11.02	2.05
10	Premium valve oil	.....	0.091	.....	1.82	.....	2.90
					1.014	6.19	0.97

is common in marine pratice, and which are then lubricated, refuse to run quietly thereafter without oil and plenty of it.

7 There is no doubt that much of the expense of cylinder lubrication and much of the annoyance, interruption of service and cost of repairs due to ineffective cylinder lubrication, can be avoided by a careful selection of the oil for a particular service. It stands to reason that a heavy low-pressure piston and the massive valves of a low-pressure cylinder at a comparatively low temperature, when using the moisture-charged steam from the high-pressure cylinder, will call for a different lubricant than will the lighter but hotter parts of the initial stage.

8 For the lubrication of cylinders using superheated steam and in internal combustion engines an oil of high viscosity and flash point is required. Some of our readers are puzzled to know how such cylinders are lubricated by oils which vaporize at temperatures less than those of the working fluid. The flash points are, however, taken under atmospheric pressure, and it is evident that the oil will remain liquid and sufficiently viscous under the high pressures used, and especially upon the surfaces of the cylinder which do not attain the temperature of the gas or steam, while it would become too limpid to serve as a lubricant or would even vaporize at atmospheric pressure and the temperature of the working fluid.

9 Attempts are made to improve the effectiveness of the lubrication of gas-engine cylinders by timing the injection of the oil. It is to be expected that a given quantity of oil will go further when injected into the piston grooves on the exhaust stroke, where it has that and the succeeding induction stroke to spread over the cylinder walls before ignition, than if injected directly into the cylinder when combustion is going on, in which case it would be only an expensive fuel. The delicacy of such timing may be appreciated when it is recalled that the speed of the piston at the center of the cylinder, where the oil is ordinarily introduced, is about 1200 ft. per min., at which rate it would take a piston one foot in thickness one-twentieth of a second to pass the oil hole, not a long time to get a column of oil into motion and stop it again. I believe sone builders are introducing the oil at the ends of the stroke while the piston is dwelling on the center. It would be interesting if some of the gas-engine men would tell us of present pratice with regard to location of feed, time of injection, and what kind of lubricant gives the best results.

Dr. D. S. Jacobus. At one time at the Stevens Institute of Technology a great many tests were made on lubricating oils. Later on

when anyone sent an oil to be tested we wrote that our experience in making friction tests had convinced us that results obtained in a laboratory gave no indication of the value of an oil as a lubricant in practical service; and further, that if we made friction tests a statement as to their lack of reliability would be incorporated in the report. As a rule those wishing to have friction tests made did not then send oils to us.

2 The above decision was not reached until a great many tests had been made, with various forms of machines. Professor Denton conducted most of these tests, and it was my privilege to be associated with him in the work. In one machine a car axle was mounted on roller bearings and so arranged that the full load that would ordinarily come on the journal could be applied and measured. This machine was run at speeds corresponding to railroad practice, all the conditions of service being copied as closely as possible. The journal was given an end play and a blower was used, to blow air over the entire journal at the speed at which a car would ordinarily travel. Very accurate results were obtained with this machine, which was run for eight or ten hours a day for a number of months.

3 Tests were also made with another machine for determining the friction inside a steam engine cylinder. The frictional force exerted on the piston rings was measured in a way that eliminated the effect of inertia. Before the advent of the above machines we had spent many hours with a Thurston friction oil-testing machine in which the pressure is applied in two directions on the journal at all times, which is not the usual case in practice; and the idea in constructing the special machine was to place the oils under as near service conditions as possible.

4 One feature which was very definitely shown was, that if friction tests are to mean anything at all, the friction of one oil should be compared with the friction of another. The condition of the journal greatly affects the amount of friction, and if a journal is worked down to give the best results, it requires a long time to get it back into shape, should it accidentally become abraided. In fact, as the brasses wear away, the variation in form of the bearing surface will sometimes change the coefficient, and it is no exaggeration to say that with a given oil, results varying by 100 or 200 per cent will be given, by a journal in the same condition as far as the eye can see.

5 In comparing the results with a standard oil, we first determined the friction of the standard oil, then the friction of the oil in question, and then went back to the standard oil. If the two tests with the standard oil agreed substantially with each other we reported the per-

centages of friction, but if they did not agree we continued making experiments, first with one oil and then with the other, until there was an agreement.

6 We made a number of attempts to measure the wearing qualities, or durability, of an oil and finally concluded that there is no such thing as wearing out an oil. Any test of this sort that can be gotten up is simply a measurement of how the oil sticks to the journal and lubricates it.

7 Professor Denton eventually developed the idea of tracing out the lubricating qualities of an oil by observing how it acted in the practical field. One young man connected with the department made a number of trips across the ocean on freight steamers, to determine the quality of one oil as compared with another for marine service. Tests were also made on locomotives and other classes of engines. To me was assigned the task of determining which of two oils would be the best for service on transatlantic liners, by making a trip from New York to Liverpool on the American Line steamer New York, and I can assure you that there is much besides friction to be considered in selecting the oil which will be the best for that sort of work. I do not blame engineers for refusing to try a substitute when they obtain an oil that suits them. There is enough strain in keeping up a maximum speed in an ocean liner without having to worry about the quality of the oil. As far as I know, oil of the composition decided on is still used for the work.

CHAS. A. HAGUE. In lubricating steam cylinders, the condition of the steam makes a difference as to how the oil will hold to the surface. I know of cases in which a very good cylinder oil, recognized for many years, failed entirely to lubricate the valves and cylinders of a large Corliss engine. I think it cost $1.10 a gallon, and after a great deal of experimenting it was found that an oil which cost 30 cents a gallon and was inferior in appearance, lubricated the cylinder perfectly. Apparently the steam was so wet, and the better oil so viscous, that the valves and cylinders did not get any lubrication at all, while the cheaper oil was thinner and more easily distributed

GEORGE A. ORROK. Mr. Souther's remarks concerning brass bearings and babbitted bearings have reminded me of my own experience with bearings, not in automobile engines, but in the larger steam engines. Formerly, it was customary to make the cross-head box and the crank-box of brass, sometimes babbitted, but very rarely.

The boxes were lubricated as well as possible, but they would become hot.

2 We finally discontinued using brass, and I think every large engine builder today uses cast-iron or steel boxes and lines them with babbitt. The trouble from hot bearings has disappeared. I do not think this is due to the use of a larger amount or a better quality of oil, but to the superior construction of the boxes themselves.

PROF. P. E. WALTER.[1] It is well recognized that the usual tests for viscosity, flash temperature and acidity, are valuable for their discriminative testimony; but every engineer is looking for some conclusive test which will give positive information without entailing the expense, time and frequent loss, occasioned by a long-time trial in actual service.

2 One phase of the flash temperature which is easily overlooked, is its relation to the rate of loss by evaporation. Practically all our commercial and lubricating oils are more or less complex mixtures of hydrocarbon compounds, any given oil being made up of compounds vaporizing at different temperatures. The lightest hydrocarbon in the group fixes the flash temperature, and also influences in large measure the evaporation loss. This loss is often a serious consideration, since in continuous oiling systems the gradual thickening and deterioration serve to increase the difficulty of handling the oil, to increase the friction loss in the bearings and cause an actual loss of oil. It seems natural to expect that some relationship should exist between flash temperature and loss by evaporation.

3 With a view to such relationship, the writer carried out in his laboratory a test for evaporation loss under several different temperatures, of three oils whose flash temperatures were carefully determined. While the data thus found are insufficient to establish fully a law of relationship, the results may be expressed with a fair degree of approximation by an equation of the form

$$L = C\left(\frac{t}{100}\right)^3 (400 - T)^2$$

where L = loss in per cent of weight in 300 hours.
t = temperature of oil in use.
T = flash temperature.

For oils flashing between 325 deg. and 375 deg. fahr., C may be taken at a value of 0.0003. For oils flashing below 300 deg., C is about

[1] University of Kansas, Lawrence, Kans.

0.00075. These figures apply to oils at temperatures above 100 deg. fahr., to which oil confined in closed crank cases of engines and in some hollow bearings may be subjected for long intervals. The great difference in evaporation loss between oils flashing below 300 deg. and those flashing only 30 or 40 deg. higher, is due to the presence in the former of light hydrocarbon compounds, left there by an inferior method of refining. Such low-flash test oil can of course, be seriously considered only for intermittent service in cool places.

4 The viscosity test is a valuable one, but cannot be taken as a direct measure of the lubricating value of an oil. My own experiments, with several oils all of which are capable of bearing a given load and otherwise performing satisfactorily in service, have shown that the one having the least viscosity will give the lowest coefficient of friction. To the careful practicing engineer who is looking for the best lubricant for his work, and who has found out by the test of actual service that certain oils will operate successfully in his plant, this gives a method of making a final selection.

5 Care must be exercised in following out this method, however, since an oil might be under too great a bearing pressure and still be able to keep the bearing temperature down to a permissible point. Some form of endurance test should be made to yield information on this point. Furthermore, in making the viscosity test, each oil should be tested at several temperatures over a range including that at which the oil is used, remembering that the actual oil film in the bearing is probably several degrees warmer than can possibly be registered by any thermometer set in the metal of the bearing.

6 By plotting curves of viscosity for varying thermometers it is seen that oils with viscosities markedly different at some ordinary temperature such as 60 deg. fahr., may be virtually identical at the temperatures at which they will run in a given bearing, or even reversed in relative values. Tables of viscosity-values of oils at a common temperature are virtually worthless. As a complementary fact it may be noted, that a bearing running moderately warm is not of necessity a cause for condemning one oil in favor of a second. It may simply mean that a condition of temperature equilibrium is reached with the first at a different point from the second, and a consultation of viscosity records at the proper temperatures is necessary before a decision can be intelligently made.

7 It would be both convenient and interesting if data could be obtained by which one could calculate in advance, by some exact method based on a knowledge of the oil, the amount of work lost in

friction of lubricated bearings for different conditions. This would require the determination of the coefficient of friction on special friction machines, and the experience of the writer with three different types leads him to the conclusion that this is impossible. Conditions in practice cannot be duplicated on a special machine; or at most but a limited number of sets of conditions could be so duplicated, and absolute values of the friction coefficient thus obtained are questionable.

8 The friction machine plays an important part in the investigation of a lubricant, however, provided the bearing is always supplied with oil in the same manner. This question of supply is an extremely important one, because a vital factor in the lubricating value of an oil is its ability to spread over the surface. In a special machine, an oil which would not spread well with the system of supply and distribution adopted would give poor results in comparison with some other oil, while under different conditions of supply better results might be obtained.

9 A laboratory investigation must conform to the injunction to vary one thing at a time. The writer is of the opinion that perfect lubrication is necessary in comparing the true lubricating values of different oils, the most reliable method being immersion of the bearing in an oil bath. With a limited supply of oil imperfect lubrication may exist, and the result is neither scientifically exact or of any practical value to a user whose conditions are in any way different from those existing on the machine. There are no laws of imperfectly lubricated surfaces. These considerations bring a cross fire on the man who is striving to reduce both friction and oil bills, making it necessary for him to study the problem of application as well as the oils themselves.

10 In a perfectly lubricated bearing, the resistance to motion is due entirely to fluid friction of the lubricant. This is true whether the motion of the fluid elements be considered as that of one extremely thin layer sliding over and under its adjoining thin layers, below and above, or that of round particles or balls rolling between the bearing and the journal. In either case there is shearing action between the particles, due to the motion. Viscosity is the indication of the magnitude of this shearing force, and hence for perfectly lubricated bearings low viscosity is an index of a good lubricant. But it is plain that to be a good lubricant the oil must first possess the power to form films and maintain those films against pressure, which power is indicated by what is termed "surface tension" of the liquid.

11 The thickness of the film has a double significance, also, since the thicker the film the greater may be the irregularities in the surfaces without an actual puncture; and the thicker film gives a lower rate of motion of one sliding layer of oil over its neighboring layer, if we adopt the sliding theory of lubrication, and consequently a lesser force of resistance.

12 May we not expect then, that the two tests for strength and thickness of oil films would help in the selection of lubricants? Such tests might, in some measure, be applied to greases as well as oils. In the opinion of the writer, the reason why viscosity is not a positive indication of the lubricating value of an oil is simply that it tells but one portion of the story, while the remaining portion must be told by an investigation of oil films. Such an investigation should answer the following questions:

a Do thick films give less friction than thin films?
b Does high surface tension indicate a power to resist high pressures?
c Does high surface tension indicate a power of the oil to spread well and so give perfect lubrication?

Standard apparatus for making the determinations, and a system of numerical relationships, should also be worked out.

MALCOLM MCNAUGHTON.[1] The paper is most interesting, yet lacks one important detail of information, needful if we are to draw any conclusions from the tests. This omission is a test of oils, without graphite, of such character as would indicate their entire suitability to the stated conditions of velocity and pressure. The velocity (115 ft. per min.) and pressure (150 lb. per in.) would indicate the use of an oil of quite high viscosity. The tests did not include any such oils, excepting the cylinder oils, and these were tried at such a temperature that their viscosity was low. Because of this omission we are not able to make any comparison between the best possible lubrication by oil alone, and by oil or water mixed with graphite.

2 The effect of graphite in lubrication may be best considered if we keep in mind that the total friction of a lubricated journal is the sum of the internal friction of the lubricant itself, plus the friction due to the intermittent or continuous contact of the metallic surfaces. It is plain that ordinarily as one increases the other decreases, though

[1] Joseph Dixon Crucible Company

of course we can conceive of cases where a change in lubricant might be followed by an increase or decrease of both components. We may never be able to determine in any particular case what proportion of the total resistance is due to the viscosity of the lubricant and what to metallic contact; nor is this necessary if we know that the use of any particular oil is giving us the best obtainable economy.

3 The addition of graphite, such as was used by Professor Mabery in his tests, to an oil of 154 rated viscosity, even up to 1 per cent by weight, caused no appreciable change in viscosity. A pipette of 100 cubic centimeter capacity allowed the oil, with and without graphite, to flow through in exactly 151 seconds in every trial. Therefore if there is no change in the viscosity of the oil due to graphite, any reduction of total friction must be found in the reduction of the friction between the metallic surfaces. That interposing of graphite between frictional surfaces reduces the friction due to metallic contact is so well known as to need no demonstration.

4 From the preceding statements it naturally follows that where the metallic frictional component is relatively large the graphite effect will be large, and on the other hand where it is small the effect of the graphite will be relatively small.

5 The light oils selected by Professor Mabery for his tests were well suited to show the action of graphite in reducing metallic friction though we are left in the dark as to the effect if oils of suitable viscosity had been used. Keeping in mind what has been said, the lubricating value of a mixture of graphite and water is not remarkable; for as the graphite reduces the metallic friction, the ratio of the fluid friction to the total friction is greatly increased, and the total friction lessened. Many will take exception to the statement that water is "completely devoid of lubricating qualities." It lacks but one quality of the ideal lubricant, and that is ability to keep the metallic surfaces apart. When this defect is provided against by pressure, water gives most perfect results. The matter in Professor Mabery's paper does not give us any basis for forming an opinion as to the relative merits of lubrication by water and graphite, as compared with lubrication by oil alone; to make such a comparison we must know not only their relative efficiency as lubricants but their relative cost, in order to determine their relative economy. If the water plus graphite costs more per gallon than the oil alone, and lubricates no more efficiently, then of course it becomes less economical in use.

6 We may now turn to a consideration of the way in which

graphite serves to reduce journal friction. For purposes of illustration let us consider the case of a fast running journal and a well fitting bearing in which the projection of any irregularity from the normal surface is less than the dimensions of the particle of graphite. The film of oil has its greatest velocity at the surface of the revolving journal and least at the surface of the bearing. Any particle free to move in the oil will be thrown outward and finally come in contact with the surface of the bearing. The journal's motion is always slightly eccentric so that in time every part of the bearing surface has been in contact with the journal. These two forces serve in time to produce on the bearing a veneer of graphite more or less perfect. But in case the irregularities of the frictional surfaces are greater in depth than the dimensions of the graphite particle, it must be seen that the graphite veneer may not be formed at all or be so long delayed in formation as to allow serious trouble to develop. Even approximately perfect bearings are hard to secure and the more they depart from a perfect condition the greater the difficulty in getting the true graphite surface.

7 It is clear that the rougher and more irregular the surface of the bearing the greater the size of the particle should be, in order to prevent the high points from coming in contact with the journal. This fact appears to limit the use of this very finely divided graphite to bearings of the most perfect character and which are the least likely to give trouble. Professor Mabery calls particular attention to the importance of maintaining a high state of perfection in all bearings, which of course is entirely true. But in fact, the normal condition of the average bearing is quite otherwise, and this is the type of bearing which we have most to consider. The statement that the artificial graphite is more unctuous than the natural graphite is denied absolutely. All high grade graphite is unctuous, whether made by nature or artificially.

8 This characteristic is usually determined by rubbing between the finger tips. On trial it will be noticed that while good natural crystalline graphite appears unctuous without pressure, the finely ground artificial graphite must be rubbed with some pressure and brought to a polish before the same effect is secured. The smoothness of this surface does not compare with that of every particle of crystalline graphite, while it is much less resistant to wear.

T. C. THOMSEN.[1] Professor Mabery quotes the opinon of well-known experimenters on oil-testing machines, that "such experiments do not afford results comparable with those in actual practice." This is also the conclusion to which the writer has arrived after having carried out a great number of comparative friction tests on machinery of the most varied description.

2 As Archbutt suggests, it is the quality of "oiliness" or "greasiness" that is so very important when judging the lubricating value of an oil; and this characteristic, viz., the adhesiveness of an oil to different bearing metals, is a factor that can never be, or at least has not been up to now, brought under the control of the laboratory. Two oils of practically the same gravity, viscosity, flash point, fire point and color, and both pure mineral, but otherwise made from different crude oils or by different methods, may in actual practice show a difference in friction of as much as 14 per cent. This has been recently proved in Germany, the test being carried out by the electro-technical department of the Bayerische Landesgewerbe-Anstalt in a textile mill.

3 Professor Mabery, while admitting that the conditions on the friction testing machine should be as nearly as possible the same as the conditions under which the oil is actually to be used, has carried out his experiments on a machine which to my mind is as far away as possible from being representative of any bearings at present employed in any kind of machinery. The diameter of the testing machine shaft is 1 in., and the length of the journal about 12 in. In order to keep this bearing and journal in satisfactory operation Professor Mabery found it necessary, not only to mill the journal and bearing to mechanically true surfaces, but by repeated careful milling an even higher degree of permanent evenness had to be maintained. Also, bronze was found unsuitable as a bearing metal and white metal was adopted. (The explanation of this is no doubt that for the abnormal length of the bearing in proportion to the diameter, white metal being more yielding than bronze produced a more uniform bearing pressure.) Further, means were provided for examining the surface of the journal, which means that the oil film was broken, making it difficult for the oil to perform its function, namely that of forming a separating film between the journal and the bearing. If finally the sides of the brasses were not eased away (chamfered), this would form another condition of the bearing which is not, or should not be, met in good engineering practice.

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4 After Professor Mabery has succeeded in making his most exceptional bearing work satisfactorily, he concludes that the same amount of care should be applied in ordinary factory operations. This suggestion is, to say the least, absurd, as it would increase the cost of all machinery several times and it would not be possible to maintain such a standard of accuracy for any length of time

5 The net result of Professor Mabery's tests is, that when graphite has been used and the supply of lubricant cut off, it will take a longer time for journal and bearing to "seize" than if no graphite had been employed. This is easy to understand, and has been known ever since graphite was first employed as a "lubricant."

6 Wherever journals or bearings, or say the metal of steam-engine cylinders, are of a porous nature, graphite will fill the pores, acting as a leveller of the surface. If a bearing is cracked and it is desired to continue operation, as otherwise a decrease in the works output might result, graphite can be used to fill the crack, preventing the oil from escaping and thus making it possible to keep the machinery operating the desired length of time.

7 As to the actual friction of bearings in good condition, whether graphite be used or not the friction is practically the same, as shown by Professor Mabery's experiments.

8 Referring to Fig. 2, the coefficients of friction with Nos. 2, 3 and 4 are practically the same. That it is possible to use fuel oil, kerosene and water will be explained by the low bearing pressure and exceedingly good finish of the journal and bearing.

9 It will be understood that a bearing pressure of 70 lb. per sq. in. on this particular bearing, due to its better finish, can be more easily sustained than the same pressure in an ordinary bearing. In other words, if the same mixtures of graphite, fuel oil, kerosene oil or water, were used on ordinary bearings, they might not be able to sustain more than 15 to 20 lb. per sq. in. It will be noticed that Fig. 3 does not contain any curve for water and graphite alone, probably because even with this low pressure it was not possible to operate the bearing without trouble.

10 Referring to Fig. 4, this shows that with an oil consumption of 6 drops per min. it was not possible to maintain an unbroken film. This shows either that the oil must have been of poor quality or that the edges of the bearing were not chamfered, as being sharp they would scrape off the oil film.

11 The differences in the coefficient of friction as shown by the different curves, Nos. 2, 3 and 4, are not greater than might be antici-

pated on any friction machine, and do not prove that the admixture of graphite reduces the coefficient of friction. What the curves do show is that with graphite, after the oil supply has been cut off, it takes a longer time for the friction to increase rapidly than when oil alone is used.

12 As to Figs. 6, 7 and 8, the conditions under which these cylinder oils are used in actual practice, with the steam engine piston moving to and fro over a film of oil mixed with water from the steam, and the oil more or less emulsifying with the water and being heated to a temperature much higher than 210 deg. fahr., are so different from the conditions in the testing machine, that for the purpose of comparing the lubricating qualities of the different oils they have no value. Further the bearing pressure in steam engines between the piston rings and the cylinder walls should always be very slight, nothing like 1200 lb. per sq. in. The temperature of the experiments is stated as 210 deg. fahr., but what is most important to know (from Professor Mabery's point of view) is the actual temperature of the oil film. This I take it, it has not been possible to obtain, and a slight difference in temperature of the oil film would make a very considerable difference in the viscosity of the oils (being cylinder oils), which again would have a considerable influence on the coefficient of friction.

13 Judging from Figs. 7 and 8, it looks as if the trials with the straight oils (600W and Galena cylinder oils) had not been carried out under the same temperature of the oil film as the trials with the oil mixed with graphite. For instance, in Fig. 7, if the straight oil had been a little higher in temperature the coefficient of friction for the first portion of the diagram, up to the point of oil cut-off, would have been lower, and after that point, the oil being warmer, it would be squeezed out of the bearing more quickly, with a resultant sharper rise in the curve for the coefficient of friction, thus getting closer to the curve for the graphite and oil mixture. The same remarks apply to Fig 8. This suggests that the coefficient of friction, if the temperature of the oil film had been the same in every case, probably would have given the same diagram for the oils used straight as for the oils mixed with graphite.

14 In conclusion, as to the practical question of using oils mixed with graphite, Professor Mabery says that it is possible for graphite in the deflocculated condition to distribute itself readily in water, but that it is quickly precipitated by impurities. As it is not possible for any practical purpose, except perhaps for certain special conditions, to apply a mixture of water and graphite, it will be necessary, in order

to carry out Professor Mabery's suggestion, to use a mixture of oil and graphite, and I presume that also in this case slight imparities would cause the graphite to be precipitated. Therefore if it were used in oil cups or lubricators, without mechanical continuous mixing, the graphite would precipitate and choke the oil channels leading to the bearings. Further it is the usual practice in many engineering works, to filter large quantities of oil, for re-use, after being used on bearings. It is well known that waste oil mixed with graphite quickly clogs up the filter pads and makes them inefficient, necessitating frequent cleaning and recharging with filter material.

15 As to oil circulating systems, where a certain quantity of oil is continuously circulated through the bearings, returning to a filter and passing an oil pump, and again forced out to the bearings, in such a system also any admixture of graphite would in time mean accumulations of deposit in the lubricating pipes, eventually putting the system out of action if not frequently cleaned out.

The Author. The interesting scientific and practical observations accumulated in Dr. Conradson's extended experience in handling lubricants, and here placed on record, will be extremely useful as representing the present state of knowledge on this subject. His results from the testing of oils on the larger machine indicate what I have noted in a great variety of lubricants, that the coefficient of friction very materially diminishes with increasing pressures.

2 Dr. Conradson's tabulated data contain much practical information. Table 1 demonstrates the deterioration from long continued use, that every oil must undergo, by evaporation with consequent changes in congealing point and in viscosity, and lessened durability. His method of testing under constant temperature affords an accurate measurement of the total energy of friction. The total heat absorbed by the water used in cooling may readily be ascertained.

3 As Mr. Davis suggests, temperature tests on factory bearings may give valuable indications as to the comparative efficiency of oils. Why may not some form of friction indicator be set up in any factory for observations on friction?

4 It takes a good oil, as Mr. Souther explains, to stand the hard usage on automobile bearings, especially in crank-case lubrication; decomposition to a greater or less extent is sure, with deterioration in lubricating quality, even to carbonization at increased temperatures

5 The data of comparative cost of lubrication, tabulated by Mr

Low, illustrate forcibly the great waste that may follow from careless selection of oils, but more especially from lack of attention to the proper use of lubricants.

6 The suggestion by Dr. Jacobus that a standard oil be selected for control of the condition of bearings, seems to be about the only way whereby these conditions may be made dependable for accurate work; in long-continued tests an operator can decide quite accurately from experience, with occasional use of this aid.

7 Mr. Orrok's experience with brass bearings appears to indicate as I have observed on the experimental scale, that babbitt adapts itself more readily to the condition of the journal, especially in heavy work, with less heating and less friction, perhaps by what may be termed metallic flow.

8 As Mr. Walter remarks, continued use of an oil, especially with considerable agitation as in flooded bearings or in automobile-crank cases, causes evaporation of the constituents, with consequent thickening and gumming, it may be to carbonization with high temperatures. His experience with reference to viscosity coincides with my observations, that economic selection and use of an oil must depend on the intelligence of the engineer for proper appreciation of the data of chemical, physical, and frictional tests, and ability to adapt them to particular conditions. His lucid explanation of the manner in which the film operates, in relation to viscosity and surface tension, should, I think, include the important influence of oiliness. In answer to his question *a* I would suggest that the thinner the film the less the internal viscosity, and the lower the coefficient of friction. I believe this holds true in a large number of observations, extending through the wide range of oils from kerosene to the heavier cylinder oils and greases.

9 To answer fully the several pertinent questions suggested by Mr. McNaughton would require more space than is available. The speed in these tests is not 115 ft. per min., as he infers, but the 450 revolutions are equivalent to about 400 ft. per min., and the viscosity of the great number of automobile oils examined during the past year is less than 300 sec. Saybolt at 70 deg. The viscosity of the particular oil shown in the chart is 196 sec. Saybolt, at 70 deg. I did not include the data collected on all these oils, as it would have extended the paper to an undesirable length. I may state, however, that every oil examined gave a higher coefficeint alone than when mixed with 0.35 per cent graphite; more than this amount is not desirable, as I explained.

10 I think Mr. McNaughton will find an answer to his question, as to the relative efficiency of water and graphite, as compared with oil, alone, in Fig. 2, which gives the coefficient of an automobile oil alone as nearly 0.02, and that of water and graphite as 0.01; for as mentioned above, in every oil examined graphite diminishes the coefficient.

11 As to the question of comparative cost, I think it is a fact that lubrication by water and graphite is considerably less expensive than lubrication with oils; certainly this is true for automobile lubricants that cost the consumer 50 cents to $1 per gal.

12 Referring to the question of the detailed action and movement of the graphite particles on the journal and bearing, it is a distinguishing quality of deflocculated graphite that it readily forms a continuous coherent film, and serves especially as a surface-evener, by filling up irregularities of the frictional surfaces, so that they approach approximately the condition of perfect bearings. I have recently seen striking examples of this action on journals and bearings run for many months continuously, with lubricants containing the regulation proportion of graphite. A much smaller amount of oil with graphite, and of the oil alone, is required to support the same load and speed after this long-continued use of the graphite. This seems to be an important function of the graphite.

13 I must take issue with Mr. McNaughton as to the comparative specific qualities of deflocculated graphite, which differentiate it from the other forms and adapt it especially for lubrication.

14 Referring to Mr. Thomsen's strictures on the Carpenter machine and my manipulation of it, I would remind him that I suggested the adaptation of a machine to the particular conditions of factory operation. It is not true, as he asserts, that the friction is the same whether oil alone or oil with graphite be used. In every oil I have tried, deflocculated graphite reduces the friction, as is shown on the charts accompanying the paper. The "net results" of my tests are, reduced friction, and longer life of a lubricant carrying deflocculated graphite. This systematic and dependable use of graphite has been known since it has been used in deflocculated form.

15 Figs. 6, 7 and 8 do show plainly that graphite reduces friction in heavy oils at steam temperatures, and that a film of graphite forms and supports the bearing for a long time after the oil is shut off. In Fig. 7 and Fig. 8 the temperatures of the oil films may have been slightly lower than that of the bearing, but not much; at any rate the

temperature of the oil alone, and that of the oil with graphite, were the same.

16 Mr. Thomsen's allusion to my statement that graphite is quickly precipitated from water by impurities leads me to explain that this statement was intended to apply to the deflocculation of the graphite by alkalies and acids, or in general by electrolytes. It does not apply to the ordinary impurities in lubrication either by water and graphite or by oils containing graphite.

17 Mr. Thomsen's statement, that "it is well known that waste oil mixed with graphite quickly clogs up the filter pads and makes them inefficient," shows that he has yet to learn that deflocculated graphite filters readily through any medium. But this use of the mixtures which I have tested, dependent on filtration, is not contemplated in their proper economic application.

CAST-IRON FITTINGS FOR SUPERHEATED STEAM

THREE PAPERS BY PROF. IRA N. HOLLIS, PROF. EDW. F. MILLER AND ARTHUR S. MANN, PUBLISHED IN THE JOURNAL FOR DECEMBER 1909.

ABSTRACT OF PAPERS

The paper by Professor Hollis stated that the examination of cast-iron fittings after long exposure to superheated steam considerably above 500 deg. in temperature has disclosed distortion, cracks and permanent change of shape. Test pieces taken from such fittings have shown irregularity of strength and apparently some reduction. The case has not been conclusively made out by exhaustive tests. The theories as to chemical change rest on doubtful grounds.

Three large cast-iron fittings, two of which had been exposed in service longer than one year, to superheated steam of 578 deg., were burst by hydraulic pressure and found to be amply strong for all purposes, although the test pieces subsequently cut from them failed fully ten per cent below what should have been the tensile strength of air-furnace gun iron out of which they were made. No tests were made of the original metal and nothing conclusive was proved except that the bursting strength of a 14-in. T was 1650 lb. per sq.in.

A calculation of the stresses set up by the expansion of the long line of pipe in which the T's were placed gives a possible stress of nearly 4000 lb., and even more, due to expansion exclusive of the stresses imposed by the steam pressure. It would seem therefore that the deterioration, if any existed, was due to the absence of expansion joints.

The paper by Professor Miller gave the results of tensile tests as specimens of cast iron, gun iron and steel, certain of which were first subjected to the action of superheated steam. While the tests were so few as not to justify many conclusions, it was evident that the metals had suffered a loss in strength due to their exposure to the steam.

The paper by Mr. Mann stated that there is a growing feeling among engineers that nothing but steel will answer as a container for highly superheated steam. A sound high-grade steel casting is practically unaffected by any reasonable superheat and if the user is willing to pay ten cents or twelve cents per pound he need fear no trouble from steel fittings and steel valves. Steel at four cents or five cents per pound has not always proven a success and a reliable material of reasonable price is desirable for a great deal of steam pipe work. A high grade cast iron is capable of withstanding the demands made upon it by superheated steam, and fittings made of this material have been in successful use for the past five years. The paper describes the experience with these fittings.

B. R. T. Collins. Last summer I ran across three valves on pipe lines from boilers on the main steam header so located as to be subjected to excessive expansion strains as described by Professor Hollis. They were 10 in. extra heavy valves, with ribs running between the end flanges and also between the bonnet flange and end flanges. These latter ribs were cracked from 1 in. to $1\frac{1}{2}$ in. deep on all three valves. In addition one valve had a crack 1 in. deep in one of the longitudinal ribs, and in one place on the body showed small criss-cross cracks when examined with the miscroscope.

2 The face-to-face length of this valve was originally 18 in. but after two and one-half years' exposure to a superheat of 150 deg. this had increased to $18\frac{7}{32}$ in. This valve was removed, broken up, and pieces sent to Professor Miller for testing, which showed a tensile strength of 11,300 lb. per sq. in. The iron was very coarse, with crystals something like $\frac{1}{8}$ in. across. This valve evidently was of very poor material to start with, or else it was seriously affected by expansion strains due to its location or to the superheat. Probably all three of these conditions had their share in producing the result obtained.

George A. Orrok. When we first considered the use of superheated steam in our power stations a few years ago there had been developed a type of steam piping which most engineers considered excellent. The piping itself was of steel with VanStone flanges, the flanges being of sufficient thicknesses to prevent buckling. The fittings were all cast iron of a carefully worked out pattern, much stronger than the ordinary high-pressure fittings. The valves were of similar design and the whole piping system was bolted together with steel bolts of larger size and greater number than the ordinary extra heavy standard required.

2 This piping system gave absolutely no trouble with saturated steam. The up-keep of such a system under power station conditions with 200 lb. steam pressure over a period of a number of years was almost nothing; in fact less than $100 was spent on one pipe line in about three years time.

3 Superheated steam, however, introduced another factor, and a very important one. From certain tests made by the General Electric Company it was considered that this superheat might vary over a range of more than 200 deg. and the temperature strains brought upon the piping and the valves would be severe. It was finally determined to make the entire pipe line of steel. The prices on

steel valves and fittings were only a little higher than if of a good quality of cast iron of the thickness required for the high pressure and excessive temperature strains. We adhered to the steel piping with the VanStone joint, but made the VanStone flange of cast steel from the cast iron pattern. The steel fittings were not as heavy as cast iron ones of the same size but differed considerably in the detail of design. The steel valves followed the design of the fittings and were of various makes, both single and double wedge.

4 Our experience with the steel valves has been good and we feel that they are giving better satisfaction than was to be expected under the circumstances. Troubles developed from blowholes, however, which led to an investigation of the subject about a year and a half ago. We traced most of the blowhole difficulties to improper moulding, improper gating and to over-oxidized metal in the case of Bessemer steel and cold metal in the case of open-hearth steel. The valves and fittings are about equally divided between Bessemer and open-hearth steel, all of the former being made, however, on the baby converter by the Tropenas and Zenzes process. The manufacturers understand better today how to handle the work, and the castings which we are receiving are much better than they were two or three years ago. I believe our troubles with the steam lines resulting from superheat are now practically over, and on one steam main in particular we have done nothing in a year and a half. Whether or not the valves can be shut off absolutely tight I do not know as we have had no reason for doing this during the time.

5 After our first installation of steel fittings and valves I had occasion to look up a number of power stations in which cast iron fittings and valves had been installed for use with superheated steam. In one of these stations I saw fittings which had been under the action of superheat for approximately nine months and had been removed because of the many leaks which had developed. The castings were supposed to have been made from the best air furnace iron, but were swollen and bulged practically all over, the outside being covered with fine hair cracks. None of the castings had gone to pieces but practically all had developed leaks, and were being replaced with steel.

6 At another station a cast iron valve had gone to pieces causing quite a little damage and many other valves and fittings had been seriously affected. There were a number of vertical engines in this station in which superheated steam had been used. All the high-pressure cylinders had cracked in two or three places and they were

replacing the cylinders and had so arranged their pipe line that no more superheated steam could get to them. The fittings which I examined, taken from the superheat line had all undergone a growth in size and the outside was covered with fine hair cracks and seemed very much swollen. Analyses of the metal showed a silicon content of from 1.88 per cent to 2.33 per cent, phosphorus about 0.7 of 1 per cent, low manganese and almost no combined carbon. The tensile strength of the material after its exposure to superheat was in the case of the iron with the silicon content of 1.88 per cent about 4500 lb. per sq. in.; in the case of the silicon content of 2.33 per cent it averaged about 8500 lb. per sq. in. We have no means of knowing what this was when it was first made. Microphotographs of the etched surfaces of this metal show the essentially open character of the iron. In this particular station the superheaters have been removed and their troubles have ceased.

7 In view of the many and excessive strains likely to come on a pipe main with 200 lb. pressure and more or less superheat I have not felt that we are justified in installing cast iron valves and fittings. Even with saturated steam at the above pressure and with the length and size of mains which we are using today in our modern stations it seems to me that the extra expense for steel is justifiable and might probably be saved many times over in the cost of up-keep during the life of the station.

8 A few years ago it was the general impression that superheated steam could not exist in the presence of water. This statement has been made many times and no longer ago than at the Annual Meeting. That this idea is fallacious is, I think, the generally accepted belief today, and we have good evidence that it is possible in a steam pipe carrying steam at 200 lb. pressure and 200 deg. superheat to have a stream of water flowing along the bottom of the pipe. In this case the bottom of the pipe would be at a temperature of possibly 380 deg. fahr., while certain other portions of the pipe in contact with the superheated steam might have a temperature between that of saturated steam and the maximum temperature of superheat.

9 Regarding the difference between European cast iron and American cast iron, it has been my impression that the pig-iron manufacturer here is always trying to make a grade of iron which will command a high price in the market. This iron must be an open iron with reasonably high silicon and almost no combined carbon, the carbon content being in the graphitic state. This iron

will sell readily. If the quality of the iron fell off, and because of a lower silicon content more of the graphite was converted into combined carbon, the iron would become harder—more difficult to machine—and would not command as ready a sale. In Europe, it is my impression that they make much harder iron and are willing to spend the money to machine it. In America we demand an open iron that can be machined easily.

10 If Mr. Mann continues his researches and considers his test specimens in the light of the volumetric composition of the iron; that is, the volume which the compounds of iron and silicon and of iron and carbon occupy in the cast iron, in comparison with the volume occupied by the iron itself, he may find some interesting results.

11 Referring to air-furnace iron, or gun iron as it has been called, I think the great difficulty is the fact that it is almost impossible to control the regular composition of the product. The reverberatory furnace, while a comparatively simple piece of apparatus, is remarkably delicate, and uniform results are obtained only when the very best of care is taken. It is a comparatively easy thing to refine high silicon iron to some kind of refined iron, but it is a much harder thing to get a uniform result from each heat.

W. K. MITCHELL. The following notes are taken from several years' personal experience with superheated steam and its effect on cast-iron valves, fittings, etc.

2 Our first intimation that cast-iron fittings and valves gave trouble under superheated steam conditions occurred about three years ago and came in the nature of a surprise, as we had been using superheated steam for some years previous.

3 The first case was in a railway power plant for a high-speed electric line. The plant had been running for several months under a fairly constant load, but owing to a falling off in traffic it was decided to cut down the service to one-half or less, which made the load quite variable. Three months after this had been done the trouble with the fittings and valves began to develop. It was first found that the valves could not be closed tight, and gaskets were giving trouble. Then fittings began to show signs of weakness, cracks appearing on the outer surface.

4 Fortunately these cracks never extended through. In an 8 in. by 6 in. double tee, the metal of which was about $\frac{7}{8}$ in. thick, the cracks did not extend more than half way through, which would

indicate that there is no advantage in very thick castings under such conditions. The most serious of these cracks occurred at the junction of the flange and fittings, and kept growing to so alarming an extent that several fittings were replaced. It was then noticed that the old fittings had lengthened considerably. The original length of some 8 in. by 6 in. double-tee fittings was 35 in., and when taken out and cooled they measured $35\frac{5}{8}$ in. to $35\frac{3}{4}$ in. They had been in service about nine months. Open hearth cast-steel fittings and valves were substituted for those of cast-iron materials and have been working satisfactorily ever since.

5 About the only information we could get bearing on the cause of this growth was from a paper by A. E. Outerbridge, read before the mining and metallurgical section of the Franklin Institute in January 1904. Mr. Outerbridge stated that by repeated heating and cooling of bars he had caused the metal to grow to an almost incredible extent. He exhibited a test bar, the original dimensions of which were 1 in. square cross-section and 14 $\frac{13}{16}$ in. long, which had been heated some 27 times to a temperature of about 1450 deg. fahr., and cooled again by various methods, some slow and some fast, until at the end of the treatment it had grown to a length of $16\frac{1}{2}$ in. and a cross-section of $1\frac{1}{8}$ in. square. The similarity between the action of the fittings above referred to and the test bars which Mr. Outerbridge exhibited caused us to investigate further along similar lines.

6 The railway plant was designed for a steam pressure of 175 lb. per sq. in. and superheat was intended to be 150 deg. fahr. That this temperature had been greatly exceeded, however, was made evident by the discovery of a board that had been charred by contact with the steam trap which rested on it. This trap was connected to a drip pipe running from one of the elbows next to the strainer on a steam turbine and was about 10 ft. below the elbow. Investigation showed that the trap had been so hot that its legs had burned holes through the board until the trap was not resting on the board at all but was suspended by the pipe.

7 The president of the company that installed the superheaters said that while they were built to give an average of 150 deg. superheat, "the real question was not one of the amount of superheat but of velocity." This seems reasonable when one considers that if the load should fall very low, the velocity of the steam through the superheaters would be considerably reduced and its temperature correspondingly raised. Again, a sudden increase in the demand for steam

would result in a rapid flow through the superheater, and steam at much lower temperature, and these recurring changes of temperature must necessarily cause rapid changes in the lines due to expansion and contraction. We therefore concluded that in this particular plant, at least, the damage to the fittings and valves was not caused by the high temperature itself, but by the constantly changing temperatures due to the change of load.

8 Our contention that the damage was due to variable temperatures seemed to be borne out by the fact that in a cotton mill plant installed three years previously, where the steam requirements for pressure and superheat were higher than those mentioned (the pressure being 200 lb. per sq. in., and superheat 200 to 250 deg. fahr.), the fittings and valves were of regular cast iron, and there had been no trouble to speak of. The load was practically constant, however, varying not more than 15 per cent at any time.

9 On account of the discussion in several publications in the spring of 1908 regarding the disastrous effects of superheat on cast iron, the owners of the mill grew anxious about their piping and asked the writer to look over the system. He found everything normal; the fittings were tight, valves could be operated freely, and in a general way the plant was in good condition. The first installation had been made in 1903, and a second one in 1906, using the same class of fittings and valves.

10 The writer suggested that measurements be taken of all the fittings and valves in the plant and records kept of changes. The original dimensions were determined as closely as possible from the patterns and records of construction, and beginning with July 1908, records were kept of the dimensions of the fittings for a period of nine months. Although the changes in the dimensions were slight, the increase in length of certain of the fittings was such that it was thought unsafe to continue them in use and steel fittings were substituted throughout. Most of the valves, however, are still in service and there have been no failures in either fittings or valves. The following will give an idea of the changes which occurred from the dates of installation to the last date given:—

A 12 in. by 10 in. by 8 in. by 6 in. cross installed in 1903 measured 24. in. in length, and in March 1909 measured $24\frac{1}{4}$ in.

A 10 in. by 8 in. by 8 in. tee increased in the same time from 24 in. to $24\frac{5}{16}$ in. and

A 12 in. by 12 in. by 8 in. by 6 in. cross, 20 in. long when installed in 1903, was $20\frac{3}{4}$ in. long in March 1909.

A 12 in. by 10 in. by 8 in. by 6 in. special cross, 50 in. long when installed in 1906, had grown to $50\frac{1}{2}$ in. in 1909, also

A 10 in. by 8 in. by 8 in. tee, $24\frac{1}{2}$ in. long, measured $24\frac{7}{8}$ in. in March 1909.

11 These facts seem to show that even at high steam temperatures, if cast iron can be kept at a uniform temperature and not cooled off too frequently or too rapidly, it will meet the requirements of superheated steam for a long period; but if the temperature is subject to frequent changes such as occurred in the railway plant, the cast iron will become disintegrated and ultimately fail within a short period of time.

12 In another street railway power house which had been in operation for a number of years with saturated steam at 200 lb. pressure, cast-iron fittings, valves and pipe were used successfully. During 1906 fourteen new boilers were added, making a total of thirty-two. The new boilers were equipped with superheaters intended to superheat to about 50 deg. fahr. New piping was installed similar to the old, with cast-iron fittings and valves, and steel pipe with steel flanges. The fittings were unusually heavy and strong. In the original installation of this piping a white metal gasket had been used, about $\frac{1}{16}$ in. thick, which was very satisfactory for saturated steam. These gaskets had a melting point of about 650 deg. fahr., and no sooner had steam been turned in from the new boilers than the gaskets began to melt, and in the course of a month or six weeks it became necessary to replace every one with material that would stand the temperature of the superheated steam. As the majority of the boilers had no superheaters it was hard to understand how sufficient superheat could be generated by the new boilers to do any harm.

13 Two years later a 16-in. tee in one of the connecting pipes between the main headers was found to be leaking. The leak becoming worse, the covering was taken off and the tee was found to be covered with small cracks or fissures similar to the cracks that had occurred in the fittings taken out of the power house first mentioned, except that a few of the fissures had worked through to the inside. The tee was replaced with a new one of the same material and dimensions. When the defective fitting was examined it proved to be something of a curiosity. Its original dimensions were 31 in. face to face by $15\frac{1}{2}$ in. centre to face. It had grown on one side to $31\frac{7}{8}$ in. and on

the other side to $32\frac{1}{2}$ in. The flanges, which were originally 25 in. in diameter had grown to $25\frac{3}{4}$ in., and as they had been bolted to steel flanges that had not changed under the superheat conditions, they had become dished to a depth of about $\frac{1}{8}$ in. The original thickness of the body of this fitting, as near as could be determined from the pattern, was about $1\frac{3}{4}$ in., but where the surface cracks were most numerous careful measuring gave a thickness of almost $2\frac{1}{4}$ in. Of course, there is always the possibility of the core moving when a casting is being made, but the thickness of this fitting was quite uniform throughout.

14 In this plant, the trouble did not stop with the fitting. Several valves began to show cracks and were replaced. Then the high-pressure cylinders of the engines became affected, and several had to be renewed. The engineers finally decided to take out the superheaters, and the plant is now running without superheat. This is a typical street railway plant, subject to changes of temperature similar to the one first mentioned. Fig. 1 shows the original dimensions of the 16-in. tee and its dimensions after coming out of the line.

15 It is of interest to note that at the same time these superheat boilers were installed in the railway plant, similar boilers with engines, piping, valves and fittings of the same type and material, were installed in a lighting plant in the same city, where practically no trouble of any kind had developed. This seems to indicate that a much more constant load is maintained.

16 It is my opinion that in plants where the load is constant and the temperature of the steam therefore constant, properly designed piping with cast-iron fittings of good material will do the work satisfactorily and be safe for a long time. I believe there is no advantage in using cast-iron alloys known as semi-steel, ferro-steel or gun iron. In the railway plant first referred to, the fittings were of cast iron from one foundry; gate valves of semi-steel from another; and stop, check and emergency stop valves also of semi-steel from a third. The results in each case were practically the same. A specimen from an 8 in. by 6 in. double tee which had been in service about nine months gave a tensile strength of 13,750 lb. per sq. in. The chemical analysis of this piece gave the following: Carbon, 2.502; Phosphorus, 0.461; Sulphur, 0.083; Silicon, 2.435.

17 Two test pieces from the 16-in. tee showed tensile strengths of 4970 lb. and 4340 lb. Chemical analysis: Silicon, 2.33; Sulphur, 0.07; Phosphorus, 0.68; Manganese, 0.39; Total Carbon, 3.18.

18 In Fig. 1 is shown the first fitting listed in Par. 10, taken from

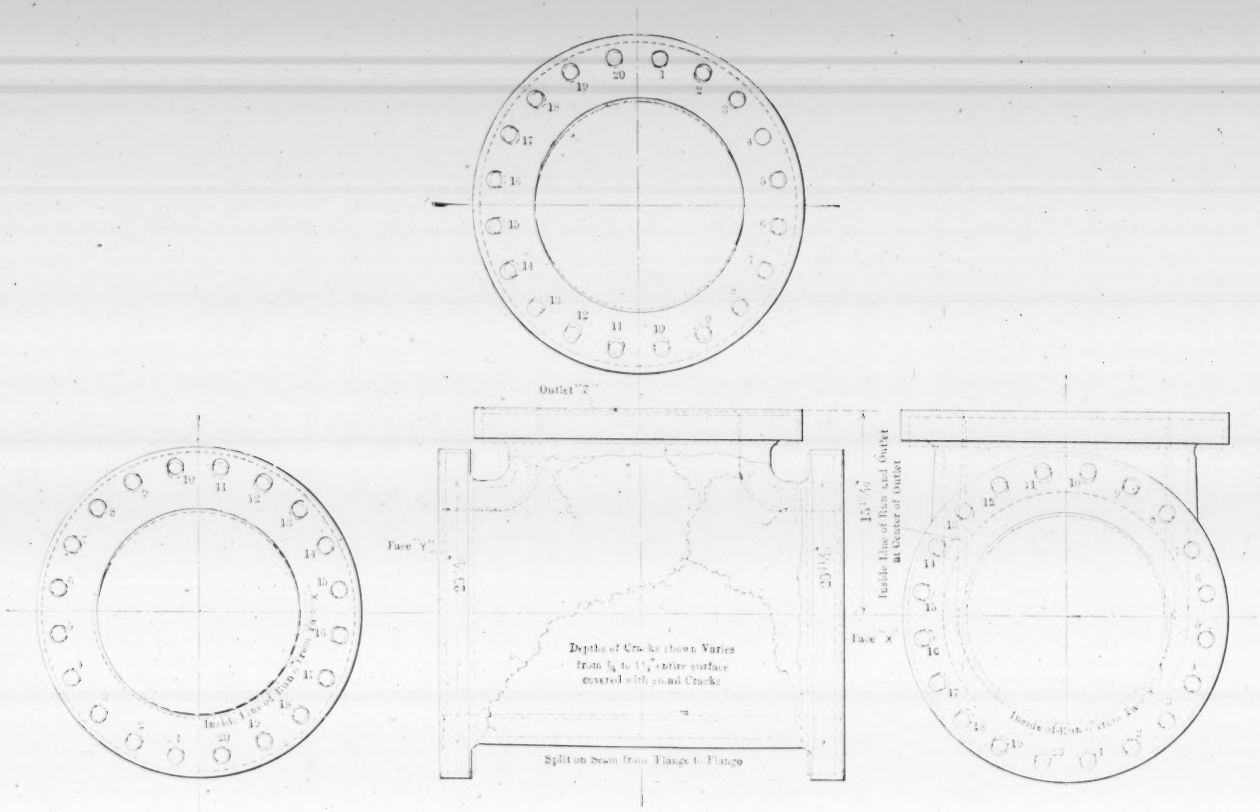
Outlet "Z"
Face "Y"
Face "X"
Depths of Cracks shown Varies from ¼ to 1½" entire surface covered with small Cracks
Split on Seam from Flange to Flange
Inside Line of Run and Outlet at Center of Outlet

FIG. 1 CHANGES IN DIMENSIONS OF 16-IN. TEE REMOVED FROM STREET RAILWAY POWER STATION

the cotton mill plant where it had been in use several years under superheated steam, the temperature of which, however, was nearly constant. This fitting was tested under hydraulic pressure. At first the pressure was put up to 1100 lb. per sq. in., when the gaskets leaked and the pressure had to be reduced to zero in order to tighten up the gaskets. The fitting was then tested again and broke at a pressure of 1250 lb. There were no serious defects in the fitting. One small surface crack was of so little moment as not to

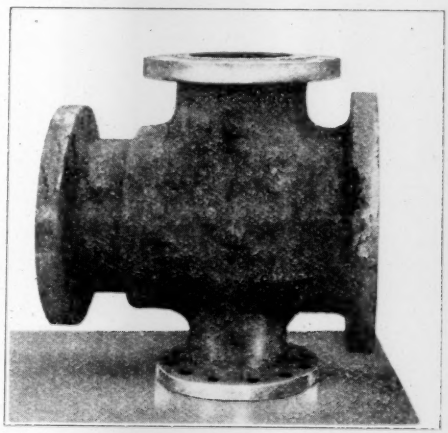

FIG. 2 12 IN. BY 10 IN. BY 8 IN. BY 6 IN. EXTRA-HEAVY CAST IRON FITTING IN USE WITH SUPERHEATED STEAM, 1903–1908

require special attention. A testpiece from the fitting showed a tensile strength of 15,900 lb. per sq. in. The chemical analysis was as follows:

Total carbon	3.05
Phosphorus	0.769
Sulphur	0.06
Silicon	2.07

19 It will be noted that the silicon in this specimen is lower than in the two castings just mentioned and I think this had a good deal to do with the case, as well as the fact that the load and temperature were constant.

20 While open-hearth steel castings seem to be successfully used under superheated steam conditions, I do not believe they will last indefinitely because of their extreme thickness. There must be

changes taking place similar to those in cast-iron, due to temperature changes, but the ductility of open-hearth steel will undoubtedly delay the process of disintegration for a longer period. The material which we recommend and use today for high-pressure superheated steam, is wrought steel throughout, with welded nozzles instead of fittings, and steel flanges, using bends in all cases in preference to short elbows.

JOHN PRIMROSE. During the past eight years the writer has been in close touch with many plants, containing upwards of fifteen hundred installations using superheated steam, and in a position where troubles would be promptly reported to him. Almost without exception these plants use cast-iron fittings in their pipe connections. The fact that no one of these plants has reported troubles with its fittings is in striking contrast to the comparatively few instances where superheated steam has been charged with being the cause of trouble with cast-iron fittings. In order that there should be no doubt about the absence of trouble due to superheat, letters were written to ten concerns known to have been passing superheated steam through cast-iron fittings for the past eight years, at from 100 to 150 deg. superheat, asking the following questions:

Question One. Are not the tees, elbows and valves of cast-iron in the branch and main steam lines leading from the boilers? Seven answered yes, two replied that some fittings and valves were of cast-iron and some of cast-steel, and one replied that while the fittings were originally of cast-iron some tees had been changed to cast-steel, but stating positively that the change was not made because of any ill effects of superheated steam.

Question Two. Are fittings of extra heavy or standard weight? Nine replied that they used extra heavy fittings, and one standard weight.

Question Three. What steam pressure do you ordinarily carry? One used steam pressure of 100 lb., six used 150 lb., one 165 lb., one 185 lb. and one 200 lb.

Question Four. Have you ever noticed any injurious effect of the superheated steam on valves or fittings? Eight answered no, one that no trouble was experienced in fittings, but that valves with cast-iron bodies and brass seats were difficult to keep tight, and one reported no trouble further than the baking of a hard deposit on inside.

Question Five. Have you ever found it necessary to replace

any of these valves or fittings with cast-steel? Eight answered that no fittings or valves had been replaced on account of superheated steam. One answered that they had replaced no fittings, but some globe valves, and one answered that they were replacing some fittings with cast-steel, but upon further inquiry it was found that this was not because of the ill effects of superheat, but because the steam mains were being changed to contain VanStone joints and they wished to change the fittings to standard length and deemed it advisable to use cast-steel.

Question Six. Of what material are the gaskets in the steam line? Seven use corrugated copper or bronze, two sheet packing, and one asbestos.

2 The chief engineer in charge of a plant in the middle west, of some 20,000 h.p., writes that nothing has developed in any of the cast-iron fittings to show that they are in any way affected by the use of superheated steam. This plant has been in operation about five years.

3 Such evidence as the foregoing proves pretty conclusively that superheated steam does not have an injurious effect on cast-iron. There seems to be no very good reason why it should. There is nothing extraordinary in the fact that several cast-iron fittings have failed when passing superheated steam. The failures were probably due to inferior metal, or to strains developed by expansion or contraction of the pipe lines, as suggested by Professor Hollis and Mr. Mann. These are much more plausible theories than that superheated steam at a temperature of 500 deg. to 600 deg. fahr. has any effect on the metal. In investigations by Mr. Outerbridge and Professors Rugan and Carpenter on the growth of cast-iron when repeatedly heated, their experiments were started at 900 deg. C. or 1652 deg. fahr. Such instances as the growth of grate bars, etc., are all at temperatures far exceeding anything used in superheated steam work for power plants.

4 Samples of cast-iron taken from fittings passing superheated steam for years have been polished and micro-photographed before and after etching, and compared with samples treated in the same way, taken from fittings passing saturated steam. The report states that there is no evidence of a change in the carbon conditions, or of exposure to superheated steam, and in support of this a well known foundryman gives his opinion that a temperature below 900 deg. fahr. would not produce any effect in cast-iron.

5 The tests of the famous Crane valve so often quoted are no proof of superheated steam being responsible for the failure. Test bars from the broken valve were compared with test bars taken from the same heat that the valve was made from, and the valve was said to have weakened. This is no real test, because castings from different parts of the same heat, or, in fact, different parts of the same casting are known to vary in strength, and it is quite likely that fittings passing saturated steam, if compared on the same basis, would be found to have suffered greatly from the effect of saturated steam! It is unquestionably true that this valve must have been subjected to other influences besides superheated steam. It is rather remarkable that the body of the valve is said to have been weakened more than the flanges—the reason given being that the metal of the body was nearer the superheated steam. Is it not more reasonable to suppose that the metal of the body weakened more than that of the flanges because it was subjected to greater fatigue on account of expansion and contraction of the pipe?

6 The writer's experience with a great number of steel fittings used for pressure parts of superheaters exposed to hot gases, has led him to conclude that the metal in steel-castings is anything but satisfactory for fittings. It unquestionably has greater tensile strength than cast-iron, which appears to be its only advantage. On the other hand, it is difficult to get steel castings sufficiently homogeneous to hold the pressure. A large percentage of castings are "doctored" before leaving the foundry, but new openings frequently develop on test after machining, and even after the castings are in place, causing the charge to be made that the castings have not been tested before sending out. This fact is further evidence of strains developing in service, other than those produced by internal pressure, which open up cavities or spongy places not discovered by shop test. Steel castings vary greatly in toughness, as shown by the great variation of elongation on test; others are so hard that machining is very difficult. While they can be bought on very careful specifications to guard against these faults, there is always the chance of porosity. The high tensile strength of the steel is not a necessity, and cast-iron made to careful specifications is amply strong. It machines well, is not porous, and can be relied on to hold the pressure.

7 Care should be exercised in the design of pipe lines to guard against straining the fittings from movement of the pipe due to expansion and contraction. Where long radius bends are the means of taking up this movement, pipe of the lightest possible weight consistent

with safety should be used, thereby lessening the force required to spring the pipe. With properly connected flanges, full weight pipe is amply strong for all ordinary working pressures, and if drawn tubing is used, even lighter metal may be adopted. In this connection the design and arrangement of the superheater is of great importance, and should be such that sudden and frequent changes in the temperature of the steam do not occur; otherwise the changes in the length of the pipe will be more frequent, resulting in a more rapid fatigue of the metal of the fittings.

8 A better way of taking care of expansion than with long radius bends, is to use ball and socket expansion joints, which have the additional advantage of reducing the amount of piping.

9 The writer agrees with Professor Hollis in charging strains due to expansion and contraction with the failure of certain fittings, and with Mr. Mann when he charges inferior fittings with the cause of failure in other cases and recommends the use of a good cast-iron containing a percentage of steel scrap for fittings passing superheated steam. This is entirely in accord with the writer's experience.

H. S. Brown believed the troubles with cast iron would be eliminated if the temperature could be kept constant, and further said:

2 I think the discussion boils down to this, that under certain conditions steel castings will give a more satisfactory performance than cast iron. The company with which I am connected has found it necessary to replace a large number of cast iron fittings with steel castings, where superheated steam was used; and the performance of these steel fittings, under the same conditions under which the cast iron fittings were working, has been satisfactory.

3 In a large number of other plants where cast iron fittings are used with saturated steam, the design of the piping is such that the stresses set up on account of expansion and contraction are very much worse than in this system; and we do not get into troubles as we did in using superheated steam. It may be that steel fittings will form a more practical and less expensive way of taking care of the conditions set up by the use of superheated steam than elaborate precautions in the way of expansion joints and the like.

E. H. Foster. Cast-iron is much too useful a metal to receive general condemnation for steam pipe fittings, whether for superheated or saturated steam, without very good reasons and the writer is firm in this opinion that such reasons have not yet been advanced.

2 Having devoted the greater part of his time for the past ten years to the study of superheated steam and the manufacture of superheaters, the writer has eagerly followed up every report of the failure of a steam pipe fitting, where superheated steam was used, and it can fairly be said that no instance has yet occurred where the weakness has not been readily explained by the poor quality of the iron, or by lack of provision for expansion and contraction without straining the metal. The many instances where cast-iron fittings are habitually subjected to steam of varying degrees of superheat up to final temperatures close to 1,000 deg. fahr. leave no doubt that good cast-iron is equal to, if not better than, any other metal for making steam fittings for superheated steam as well as for saturated steam, especially in smaller sizes.

3 In the writer's experience it is as important to have regard to the mixture of the iron to be used in cast-iron fittings as it is to secure the proper mixture for concrete work.

4 The suggestion that better provision should be made for free expansion and contraction of steam pipes, is, in my opinion, very much to the point. More care applied to this feature of the design of power plants would remove entirely from the shoulders of cast-iron the odium of being unsuitable for carrying superheated steam.

L. B. NUTTING stated that superheaters installed by his company nine years ago, and since then in constant use, have caused no trouble and have not changed their dimensions. These superheaters were made entirely of cast iron, the tubing having a smooth bore and corrugated exterior.

2 He also reported a great many superheaters installed and in operation delivering steam at a temperature of 1000 deg. fahr. On these the users have employed, without any distortion or without any evidence of weakness developing, standard makes of cast iron valves (globe valves and angle valves) under 1000 deg. final temperature. But the temperature is maintained at 1000 deg., without a variation of 25 deg. These illustrations seem to trace the cause of the trouble with cast iron fittings directly to widely fluctuating temperatures.

3 In regard to Mr. Mitchell's suggestion that provision should be made to obviate troubles from varying temperatures on cast iron Mr. Nutting asked, why not make provision to keep the temperature constant. The art of superheater construction has advanced to such a point that a uniform temperature should safely be counted

on. The plant Mr. Primrose referred to, a 20,000-h. p. boiler plant has a record of variation not exceeding 10 deg. either way from the desired amount at any time during the year, although the loads had a fluctuation of from 5000 to 35,000 kw.

ANDREW LUMSDEN.[1] In one case we have eight boilers of the Babcock & Wilcox type, equipped with heaters part of which were made and installed by the boiler company. These boilers were all connected to one 12 in. main through long radius bends, valves, tees, etc. On the superheaters installed by the boiler company there were usually 150 deg. of superheat and on the others about 90 deg.

2 When this plant had been in service about two years some of the fittings in the main were found to leak just back of the fillets and small cracks were discovered extending around one side of the tees. Some long bolts were made to go the whole length of the tees and through the end flanges, using them to make the joints. Steel tees were also ordered of the same dimensions as the cast-iron ones to replace all the fittings in the main. When the old fittings were removed, however, it was found they were from $\frac{3}{8}$ in. to $\frac{1}{2}$ in. longer than when first installed. This is a turbine station and they have had quite a little trouble with the admission valves on some of the turbines, those directly opposite the boiler carrying the 150 deg. superheat giving by far the most trouble.

3 At another plant there are boilers of the Babcock & Wilcox type and Curtis steam turbines, installed seven years ago, with a separately fired superheater on which exhaustive tests were made. The temperature of the steam leaving the superheater reached as high as 750 deg. The superheater was run for about six months and at the end of that time all the copper gaskets in the main were destroyed and the joints had to be remade. The superheater was shut down and has not been operated since.

4 The writer visited this plant a few days ago and found that no large joints had been made since the superheater was shut down. There are about ninety joints ranging from 8 to 12 in. diameter and none have leaked, but three 12-in. valves were leaking badly through the body on the under side and about the point where the seat rings are screwed in. These valves are laid on their sides and are on the boiler side of the superheater and have never had superheat in them nor in

[1] President, Lumsden and Van Stone Co., 69–71 High Street, Boston, Mass.

the writer's judgment can their troubles be due to expansion as particular care was taken to allow free movement in all the piping of the plant.

5 At another plant where they have Babcock & Wilcox boilers, Curtis turbines, etc., they have carried 150 lb. steam pressure and 150 deg. of superheat for about six years, with cast iron fittings, etc., and have had no trouble with the fittings. The valves have given them some trouble with loose seat rings and by being badly cut, and some of them have been replaced.

JOHN C. PARKER. Professor Hollis draws attention to the necessity for greater allowance for expansion in piping for superheated steam. This is important and where sufficient flexibility cannot be put into the design expansion joints should be installed. My experience accords with his statement that cast-iron fittings have been largely and successfully used for superheated steam.

2 Six or seven years ago I was called on to furnish superheaters with some of our boilers but could find none in the market to meet my ideas of what a superheater should be. A design was worked out and forty or fifty thousand horsepower of boilers have been built with these superheaters. Two plants are above ten thousand horsepower. In six years experience we have had no trouble either with cast-iron or steel fittings, valves or cylinders. I ascribe the result to the steadiness of the superheat and to the fact that condensed steam in the superheater is not intermittently carried into the steam line.

3 In one of our first installations the men started to flood the superheaters without my knowledge whenever the boilers were banked and the result taught me the effect of suddenly injecting water at 360 deg. fahr. into piping and headers which had been raised to 500 deg. fahr. Leakage started at the joints but stopped as soon as the practice was stopped.

4 I believe there is no connection between the troubles which I have been cognizant of with some designs of superheaters and the expansion of the steam mains. I believe the troubles have been due solely to fluctuations in temperature and temperature shocks caused by frequent injection of condensed steam from incorrectly designed superheaters. I recently went into a plant where a superheater had been in use for about four years. It was an independently fired superheater and the engineer had had so much trouble with piston rings in an engine with poppet valves designed especially for superheated steam, that he had cut the superheat from 600 deg. fahr. to

500 deg. fahr. and then to 400 deg. fahr. He had his ideas centered on the point at which the trouble occurred (the cylinder and rings) and would not believe that it was due to fluctuations of temperature. I finally pinned him down to the statement that "the temperature could be controlled perfectly but you had to watch it like a cat."

5 There is a superheater in the market that uses cast-iron to protect wrought iron tubes. I have seen some of these removed from boilers on account of overheating and, while the cast-iron had been red hot it had cracked less than some steam pipe fittings which had been subjected to water jets and fluctuations under 600 deg. fahr.

6 I do not think conclusions can be drawn from Professor Miller's experiments. It would require at least half a dozen tests of the same sample of cast-iron at progressively increasing temperatures and periods to obtain results of value. Ten of the tests show increased strength of cast-iron while all the steel has lost strength. One steel test (100,000 lb.) is unreasonable.

7 It is unfortunate that Mr. Mann has not given us plans of boilers and superheaters and steam piping, and such data regarding the conditions under which the fitting troubles occur, as would permit more intelligent study. I note that steel fittings have failed in some cases. This would indicate water jet action, since we have never had to renew a single steel header in our superheaters.

8 We have sixteen 800 h.p. boilers with superheaters directly over the fire running at 175 lb. pressure and up to 170 deg. superheat with no such trouble as Mr. Mann mentions in Par. 12.

ALBERT A. CARY said that after investigating a number of plants having trouble with the use of superheated steam, he had been led to the conclusion that many if not most of their troubles have been due to bad design in the piping arrangements.

2 Far greater care and better judgment is called for in designing pipe systems for superheated steam than for similar systems using saturated steam, as the strain due to expansion and contraction is greatly increased. The piping on each side of every offset should be carefully considered to see that excessive stress is not thrown upon the flanges and threads by the lever which is developed there. Several special forms of flanges which avoid the screw connection are now used to excellent advantage, with high superheat.

3 Continued flexing on one side of the flanges of fittings, due to the cooling and high degree of heating of the pipe system as steam is turned on and shut off will cause ruptures not unlike those shown in

the illustrations accompanying these papers and will be apt to change the internal structure of the metal itself.

4 As higher velocities are permissible, in pipes carrying superheated steam, than in those for saturated steam, somewhat smaller piping can be used to excellent advantage, not only decreasing the cost, but also increasing the flexibility of the pipe line.

5 Considerable care must be exercised in flexible pipe bends, which should be bent to large radii, and placed in the most effective positions with a proper anchoring of the pipe on either side of the bends. Careful supervision must be exercised in seeing that the pipe fitters make up their pipe lines in such a way as to avoid any severe springing of the pipe in order to make the flanges of the joints register properly, one against the other. Undoubtedly, neglect in these particulars of pipe design and pipe fittings is responsible for a very large percentage of the failures found in the fittings and joints in superheated steam plants.

6 I have been led to consider the handling of superheated steam in power plants under two headings, first superheated steam having a total temperature below 500 deg. fahr.; second, superheated steam above this temperature. The first-named quality of steam, with a total temperature below 500 deg., will be found better applicable to old plants, in which it may be newly introduced, and in many new plants. With such steam we may use the ordinary high-grade fittings used in saturated steam work, with some modifications.

7 The high-temperature superheated steam in power plants is more especially adapted to installations where high-pressure steam is required for steam turbines.

8 Mr. Cary referred at some length to the possibility of cast iron being affected by heat under stress, calling attention to the discoveries under the microscope of the change in physical condition of alloys produced by heat treatment, and advocated the carrying out of experiments along these lines.

W. E. Snyder. I may be able to touch upon certain particular phases of the papers and the discussion, in such a way as to contribute some of the results of actual experience covering a wide variety of conditions and several years' practice. The consideration of this subject in the discusssions seems to have broadened to include the effect of unequal heating of metal and also the designs of systems of steam piping. These matters are both directly related to the use of cast iron fittings for superheated steam, as failures may in

some cases be due to the improper design of the steam piping; also under some conditions to unequal heating of such irregular castings.

2 A common connection between engine and main steam pipe is by a branch running horizontally and at right angles to the main pipe, out directly over the high-pressure cylinder and turning down by a bend to connect with the throttle valve on top of the cylinder. When this branch is long the expansion in the steam line does not exert any harmful effect in the throttle valve, but the expansion in the branch is taken up by the change of curvature of the bend over the throttle valve, and this puts a strain directly on this valve. In two or three instances where this kind of connection was in use, the throttle valves were cracked immediately under the flange, and serious accidents narrowly averted.

3 Where the branch to the engine is short the expansion in the branch itself does not require any consideration, but the longitudinal movement of the steam main due to its expansion and contractions, transmits strains through the branch pipe directly to the throttle valve and flange on the steam chest. In one instance this resulted in a very serious accident, as the cast iron flat top of the steam chest was broken in by the expansion of the main steam pipe several feet away.

4 In another installation a 24 in. cast iron Y with an 18 in. branch split in the fork of the Y, while under 150 lb. steam pressure. Fortunately it was possible to take the line out of service before the fitting exploded, but it was a very narrow escape. All the accidents mentioned above were the direct results of the installation of systems of steam piping without proper consideration of the effects produced by expansion and contraction. All were in systems using saturated steam, and they emphasize the necessity of using great care in arranging the piping that the expansion and contraction may take place without throwing the severe strains on the cast metal members, which are always liable to failure under such conditions. Consideration of this feature of design is of still greater importance in piping systems using superheated steam, on account of the higher temperature used and the consequently greater expansion. The avoidance of expansion strain on castings in a system of steam piping is of fully as great importance as is the selection of the material from which these castings are made.

5 The effect of unequal heating of metal has been investigated by engineers in the French Navy (See Marine Boilers by Bertin &

Robertson, p. 201). The theory advanced there, which seems reasonable and is also confirmed by experience, is this: When one side of a piece of metal or a boiler tube is heated to a higher temperature than the other, the hot side tends to expand, and the expansion is resisted by the metal on the cold side. This condition puts the metal on the hot side in compression, and the metal on the cold side in tension, and if the temperature difference is great enough the metal will be strained beyond the elastic limit. When the hot side is allowed to cool it is shorter than the cold side because of the strain beyond the elastic limit which has been undergone by both sides. This results in the piece taking a permanent set, or becoming "bow shaped" away from the side that has been heated.

6 The bend away from the fire of boiler tubes in some types of boilers after they have been in service for some time, seems to be a good example of the results of unequal heating. Another example is the cracking of the large cast iron mud drums used in some types of water tube boilers. Under ordinary operating conditions, in boilers having vertical baffling, the hot gas does not come in contact with the mud drum of the boilers until it has passed at least twice across the tubes, and has thus been greatly reduced in temperature. At times, however, holes are formed in the front baffle, or through the top of the bridge wall allowing the hot gases to pass directly from the furnace to the back part of the boiler setting, where they strike the cast iron mud drum, heating it to a considerably higher temperature on the front side than on the side away from the fire. These conditions have resulted, in a number of instances, in causing the mud drum to crack perpendicular to its axis, causing serious accidents.

7 This matter of the unequal heating of metal is one of the most serious with which designers of large engine cylinders have had to contend. Features of the design of pipe fittings are very similar to those mentioned in the design of gas-engine cylinders, the irregular castings having flanges and other forms of construction which make it practically impossible to avoid having the metal considerably thicker in some places than in others. This irregularity causes internal strains in the metal when heat is applied to one side. It has been the experience of European designers of gas-engine cylinders that one of the greatest difficulties they have had to overcome is this one of distributing the metal so as to avoid the small cracks resulting from irregular expansion, which destroy the cylinder. Where trouble has occurred in the use of cast iron for large fittings in superheated-steam piping, it is possible that the experience of the gas-engine engineers will suggest the remedy.

8 As bearing upon the use of cast iron for superheated steam, particularly upon the much discussed question of the possibility of having superheated steam that is in contact with water in the boiler, an experience of the speaker may be of interest.

9 A large furnace used for heating slabs for a plate mill was equipped with a small vertical Cahall boiler for the purpose of utilizing part of the waste heat. This furnace was fired with under-feed stokers, using forced blast, so that it was possible to obtain a very high temperature both in the furnace and in the boiler, also in the stack. The boiler was set in the usual way for waste heat, i.e., with the large end down. The steam pipe was connected to a flange on the top of the boiler, this connection being made inside the conical-shaped base of the stack which rested on top of the circular boiler setting. This arrangement put a cast-iron elbow on this steam pipe, and about two or three feet of pipe on each side of the elbow in the hot gas directly over the upper drum of the boiler.

10 Tests on this furnace and boiler were continued for two weeks, observations being taken every 30 minutes. Frequently the stack temperature would rise to 1000 and 1100 deg. fahr. The thermometer on a Carpenter throttling calorimeter, connected to the steam pipe just outside the stack breeching mentioned above, ranged from 380 to 600 deg. fahr., depending on the rate of working of the furnace. This variation at times occurred very rapidly, and at the time the thermometer readings were high, the steam escaping from the calorimeter was as completely invisible as though it were natural gas. For twelve hours in succession the average superheat of the steam was 140 deg. and during the entire time the tests were being made, the superheat of each 12-hr. period averaged 120 deg. or over; the steam pressure being about 95 lb.

11 This boiler has been in operation under conditions similar to the above for at least 15 years. The cast-iron elbow and flange at the top of the boiler, although subjected to such severe service as that described, has never given any trouble. A number of other Cahall boilers using blast-furnace gas, with steam pipe connections made in the same way, have been in operation for about the same length of time, and no troubles have occurred due to the fittings. The service under blast-furnace gas conditions are not so severe as the heating-furnace installation described above, on account of the stack temperature being somewhat lower, from 700 to 900 deg. The heating-furnace conditions mentioned above are unusually severe and for that reason have been described fully. It may be added that the

boilers using blast-furnace gas-produced superheated steam, notwithstanding that the water level was only a short distance below the connection to which the calorimeter was attached. (This must not be understood as being a special feature of the Cahall boiler, as in fact it is only incidental to its operation under these conditions.)

J. S. SCHUMAKER mentioned that in the cases cited no difficulty had been experienced with the fittings of superheaters, cast iron or otherwise, but that there had been a great deal of difficulty with steam-pipe fittings. This seems to be the result of high temperature on one side of the fitting only. In the superheater itself the temperatures are balanced, to some extent at least.

DR. D. S. JACOBUS. In a large power plant that I have in mind, where the fittings are all of cast-iron, and where the superheat averages 150 deg. fahr., repeated examinations have failed to reveal any deterioration. In other cases, however, where there has been less superheat, and even where a single boiler with superheated steam has been connected into a common main with a number of other boilers furnishing saturated steam, there has been every indication that a small amount of superheat has had an injurious effect. It therefore seems that a difference in the quality of the cast-iron may affect the results, and by making a careful study of the matter and knowing the analysis of the cast-iron there is a possibility that its action under superheated steam may be predicted. In the meantime, we are furnishing cast steel fittings for all superheated steam work, as we do not know of a single case of the failure of such fittings that can be attributed to the action of the superheat.

2 The stresses due to expansion, as pointed out by Professor Hollis, may tend to produce failures. In the case of fittings broken in superheated steam lines we have found there was a stress at the point of rupture entirely apart from the stress produced by the steam pressure. In the ordinary flanges the tension of the bolts produces cross strains and the fittings give way where they would naturally fail through this strain. We have given considerable thought to the construction of flanges in which such cross strains are eliminated, but have not pushed the matter forward as we have decided to eliminate all doubt as to the safety of the fittings by employing steel castings.

3 Professor Miller's tests bear out what we have observed regarding the different results to be expected from cast-iron, as they show that although there is a general falling off in strength in one case the

cast-iron specimens did not lose in strength by being subjected to a high degree of superheat. In connection with such tests it would be interesting to investigate the action of superheat when the metals are under stress.

4 Mr. Mann's conclusion that gun iron is better than cast steel is indeed interesting, but we would not think of changing our present practice of using cast steel until gun iron is thoroughly tried out in the practical field and demonstrated all right for the work. The proper method of determining the quality of the gun metal which is used must also be developed by the necessarily slow process of observing the action of the fittings in service. It would indeed be a simple matter if bids for the fittings could be based on an analysis of the metal, and I hope Mr. Mann may be right in this belief.

PROF. H. F. RUGAN. While investigating the phenomenon of the increase in cubic dimensions of cast iron as a result of repeated heatings it became evident that the test pieces deteriorated in strength. I am of the opinion that the influences at work producing such growth at high temperatures are the same that cause the failure of cast iron fittings at lower temperatures, say at from 500 deg. fahr. to 600 deg. fahr. The effect of the higher temperatures is merely to increase the extent of the changes, producing a maximum growth per heat.

2 Further experiments to determine the length of time required to produce maximum growth developed the fact that a change in the temperature was necessary to produce continued growth. No apparent difference in growth was observed between pieces heated at the same temperature for periods of 3 hours and 17 hours respectively.

3 The test pieces were heated in cast iron muffles, carefully luted with fire clay, to protect them from contact with the furnace gases, to a temperature of from 850 deg. cent. to 950 deg. cent.

4 Experiments were made with nine iron carbon alloys (A to I) containing no graphite, the carbon content changing by 0.5 per cent from 4.03 per cent to 0.15 per cent. Other constituents were low and constant. Four bars of each alloy were cast in both sand and chill moulds. These proved to be all white irons, the samples with low carbon content being full of blow holes. No growth was observed in any save the sample A which contained 4.03 per cent carbon. This sample shrank for the first 12 heats, afterwards expanding, ultimately becoming 6.88 per cent larger than its original volume.

5 Four alloys (J to M) were also tested. Of these J, K and L were grey irons while M was a white iron. It was observed that M

followed along lines closely approximating the action of A, shrinking slightly during the early heats but growing after 12 to 19 heats had been taken; ultimately becoming 6.2 per cent larger than the original. Pieces of the bars from which the A and M test pieces were made, were inserted in the muffle to be sampled for chemical analysis after successive heats. These analyses showed that the appearance of free carbon (or temper carbon) coincided with those heats which produced growth in the test pieces. Free carbon was in this way proved to be in some way an indispensable factor in the growth of cast iron when under heat treatment.

6 The grey irons J, K and L, grew from the start, and their progress indicated a close relation between their respective growth and their silicon content.

7 To check these indications a series of alloys, with all the constituents constant save silicon, having the following analyses, were used to test the part played by silicon:

Alloy	Total Carbon	Combined Carbon	Graphite	Si.	Mang.	Sulph.	Phos.
N	3.98	0.64	3.34	1.07	0.25	0.01	0.013
O	3.98	0.68	3.30	1.79	0.23	0.01	0.013
P	3.79	0.30	3.49	2.96	0.25	0.01	0.012
Q	3.76	none	3.76	4.20	0.27	0.01	0.012
R	3.79	none	3.79	4.83	0.30	0.01	0.012
S	3.38	none	3.38	6.14	0.30	0.01	0.013

8 It will be observed that the total carbon in the series is approximately constant, that alloys N and O contain about the same amount of combined carbon, that alloy P contains about half the quantity, and that the remaining alloys contain none at all. The silicon in O and R is 0.2 per cent lower, in Q, 0.2 per cent higher than was desired. The remaining constituents are satisfactorily low and constant.

9 Test pieces N to S, measuring 6 in. by about 0.88 in., were machined from the castings. They were not taken from similar portions throughout, but haphazard, some from the gate, others from a riser either near to or at some distance from the gate. When the growth of these alloys was investigated it became evident that the locations from which a test piece had been cut had a considerable influence on the rate of expansion. It was found that specimens taken from the gate end of the casting grew more rapidly than those taken from the top of the riser.

10 In test pieces from the same part of the bar, however, these inequalities disappeared and a like growth was obtained in each alloy. A slight falling off was observed in the closer-grained irons.

11 The results obtained are plotted in Fig. 1, the coördinates being percentage of growth and number of heats. In this way the rate of growth is clearly seen. In the case of samples N, O and P, curves are plotted from the data obtained. It will be observed that the growth

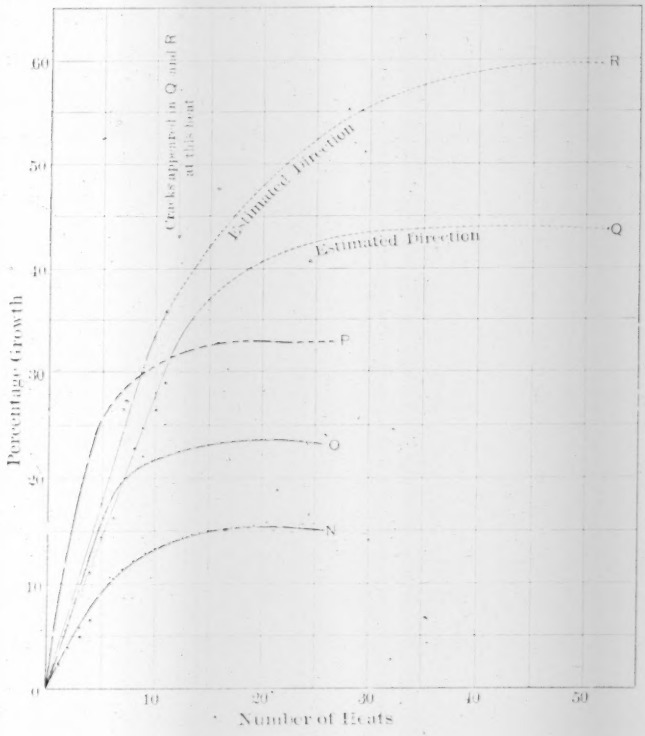

FIG. 1 CURVE SHOWING RATE OF GROWTH OF ALLOYS *N* TO *S*

is rapid at first, diminishes after about the seventh heat and stops at the sixteenth heat.

12 In the case of Q and R curves are plotted in full lines from the data obtained up to the point at which cracks appeared, viz. the twelfth heat. Beyond this the direction of the curves can only be guessed and this is indicated by dotted lines. The following table summarizes the results:

Alloy	Percentage of Silicon	Percentage Growth on Heating
N	1.07	15.40
O	1.79	23.46
P	2.96	32.85
Q	4.20	43.90
R	4.83	59.50
S	6.14	63.00

13 It is quite clear from these tests that silicon is a most important constituent of cast iron from the standpoint of growth under repeated heatings. If the ultimate growths and percentages of silicon are plotted as coördinates, the curve in Fig. 2 is obtained, which shows that, broadly speaking, the growth is proportional to the percentage of silicon.

14 To settle the question as to the influence exerted by graphite, and at the same time determine if iron-silicon alloys, containing little carbon and no graphite, would grow, three alloys (T, U and V) were experimented with, having the following analyses:

Alloy	Silicon	Carbon	Mang.	Sulp.	Phos.
T	0.65	0.17	0.17	0.045	0.017
U	1.10	0.18	0.19	0.049	0.022
V	2.71	0.19	0.20	0.051	0.033

15 Microscopic examination showed in all three cases a solid solution of iron silicide in iron. There were no traces of graphite or any other structural constituent.

16 Machined bars were heated fifteen times under the same conditions as the previous alloys. A summary of the final values is contained in the table below:

Alloy	Percentage Silicon	Percentage Change of Volume after Fifteen Heats	Percentage Change of Weights after Fifteen Heats
T	0.65	−0.025	−0.04
U	1.10	0.000	−0.03
V	2.71	+0.394	−0.02

17 It will be seen that the only alloy of the three which showed any tendency to grow was V, with 2.71 per cent of silicon. The expan-

sion, however, was very slight, and compared with that of P (2.96 per cent silicon and 3.79 per cent carbon) after the same number of heats was almost negligible, amounting to but 0.394 as compared with 31.35 per cent, the mean figure of P and PP.

18 Alloys K, N and P correspond closely to alloys T, U and V in silicon content. They also contain about 3.9 per cent of carbon,

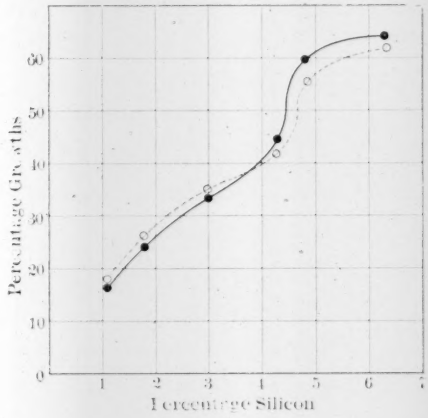

FIG. 2 CURVES ILLUSTRATING RELATION BETWEEN PERCENTAGE GROWTH AND PERCENTAGE SILICON

mostly in the form of graphite, as compared with a mean figure of 0.18 per cent carbon, none of which is present as graphite in the other series. A comparison can thus be made between the changes of volume of the two series under similar tests after fifteen heats, by means of the following:

Alloy	Carbon	Silicon	Per Cent Change in Volume	Alloy	Carbon	Silicon	Per Cent Change in Volume
T.....	0.17	0.65	−0.025	K.....	3.90	0.69	+ 5.40
U.....	0.18	1.10	0.000	N.....	3.98	1.07	+15.20
V.....	0.19	2.71	+0.394	P.....	3.97	2.96	+31.35

19 This comparison serves to emphasize anew that free carbon, even in the form of graphite, is one of the essential factors in the growth of cast irons under heat treatment. The previous series of alloys, however, N to S, brought out clearly the fact that in the constant

graphite series the growth is roughly proportional to silicon present, graphite becoming merely the agent or forming the avenues by means of which the silicon present can be acted upon. It is clear, therefore, that both graphite and silicon are involved in these changes of volume after repeated heatings.

20 A sample of test piece S, known as "S $\frac{1}{2}$," was heated to constant volume in vacuo. This resulted in a shrinkage of 0.04 per cent. The same sample was afterwards heated in the muffle to a constant volume, when a growth of 67.70 per cent was obtained. Fig. 3 shows curves plotted from these data. It will be seen that S $\frac{1}{2}$ grew rapidly during the later heats, with no cracks developed, and the sample retaining its original form throughout. S, however, grew rapidly during the earlier heats, cracks developing during the first heat, and finally breaking in two.

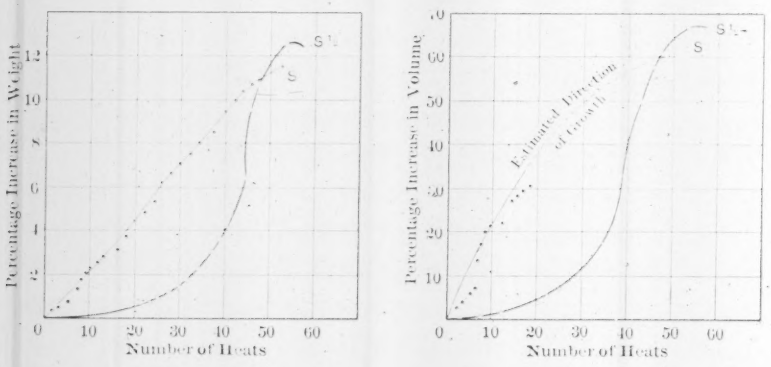

FIG. 3 CURVES SHOWING PERCENTAGE INCREASE IN WEIGHT AND VOLUME AND NUMBER OF HEATS: TEST PIECES S AND $S\frac{1}{2}$

21 Mr. A. Wolfe, superintendent of motive power of the United Railway and Electric Co., Baltimore, commenting upon the growth and final failure of some fittings in one of their power houses says, "The temperature was not constant, varying between that of the temperature of saturated steam at 175 lb. per sq. in. to superheated steam running between 500 deg. fahr. to 550 deg. fahr. total temperature."

22 From the experiments I have made there is considerable evidence indicating that gray cast iron subjected to changing temperatures from 450 deg. fahr. and up gives evidence of an oxidation of the silicon present, forming silica in a micro-crystal form, which upon

cooling causes a disintegration of the surface exposed, ranging generally along the planes formed by the graphite, changing an apparently solid wall into one showing many cracks. It is the constant recurrence of these conditions, produced by the changing temperatures, that, in time not only produces growth, but breaks down the structure of the metal.

23 The experiments conducted by Prof. E. F. Miller were so treated, cooling down each night to the temperature of saturated steam. No mention is made of any growth of these specimens. One would expect to find a marked relation between growth and loss of strength.

24 In a comparison between gray iron samples, those subjected to a heating and cooling treatment totaling 1000 hours would become weaker and larger than those kept constantly at the temperature of superheated steam for a like period. I believe that actual contact with steam is not a necessary condition in this experiment, neither is it a comparative test of the metal, which in service has one surface exposed to atmospheric conditions.

AUTHOR'S CLOSURE

PROF. IRA N. HOLLIS. The discussion of these papers brings out certain interesting and valuable conclusions which cannot fail to assist in the proper use of cast iron for parts of machinery and boilers. Previously existing differences of experience with this metal under a high temperature are shown to be due to fundamental differences of chemical composition or to variations in the temperature. From this point of view, existing data, even though conflicting, can probably be reconciled. The following conclusions as the result of the papers and discussions may be studied with profit in connection with new construction.

a Cast iron varies in its behavior under high temperature, starting from about 450 deg. fahr. In many cases it deteriorates in structure and strength to a marked degree.

b The effect of high temperature is independent of the medium producing it, whether superheated strain, hot gases or solids.

c. The change of structure or deterioration is much increased by a fluctuating temperature.

2 Where the temperature is constant, even though as high as 600 or 700 deg. fahr., the change in cast iron is not serious enough to pro-

hibit its use, but where the temperature varies considerably, the metal is certain to develop cracks and distortion that render it unsuitable for steam pipes and other parts under steam pressure.

d Cast iron of certain chemical constituents increases materially in volume when subjected to fluctuating temperatures above 500 deg. fahr.

e The chemical composition of the cast iron has a material bearing upon the change of shape and volume and upon the development of imperfections.

3 Certain facts in this connection are well shown by Professor Rugan's experiments. As he states, cast iron containing only combined carbon does not change even under high fluctuating temperatures unless the carbon begins to separate into a graphitic form. Free carbon is one of the factors assisting in the deterioration under high temperature, especially when associated with silicon. The latter seems to be the chief cause of increase in volume. Where the free carbon is constant, the growth in dimensions is roughly proportional to the percentage of silicon.

f The use of cast-iron fittings for superheated strain is inadvisable where the temperature is likely to fluctuate, but it can be safely used where the temperature is to be constant.

g Cast-iron fittings should not be placed in any parts of a steam-pipe line where there are serious bending stresses in addition to the stresses produced by internal pressure, unless the combined stresses are fully allowed for, or neutralized by expansion joints.

GAS POWER SECTION

REPORT OF STANDARDIZATION COMMITTEE

PUBLISHED IN THE JOURNAL FOR DECEMBER 1909

DISCUSSION PRESENTED AT ANNUAL MEETING 1909

DR. D. S. JACOBUS. I would ask why it is that if the engine, producer and other apparatus, are put in by the same party, it is not necessary that data shall be taken of the efficiency of the producer, as differentiated from the efficiency of the engine, etc. I may have misunderstood that feature of the report, but I remember that there was a severe criticism of a paper presented last year upon a test of a steam power plant, in which simply the fuel and the final output were measured. It was held that what was really needed was the intermediate data to show how the efficiency was obtained.

PROF. R. H. FERNALD. In Par. 24*a* of the report it is recommended that the sample be dried at 210 deg. fahr., or just below the atmospheric boiling point. The code of the American Chemical Society requires, I believe, a temperature of 105 deg. cent., or a little above boiling point, which is the Government specification at the present time.

2 One of the most difficult things is to judge the conditions of the producer bed, as the report implies (Par. 25). It is so important that a special bulletin, No. 394, has just been prepared for the technological branch of the geological survey report dealing entirely with the question of the bed. Tests of a 250-h.p. producer carried on for 60 hours continuously, showed that this length of time was not nearly long enough to give conclusive results.

3 In Par. 52 reference is made to the time required to reach the steady state in the case of large engines, which raises a query as to

the division between large and small engines. This depends on individual practice. A 250-h.p. engine might be large from one point of view and small from another.

4 In Par. 55 the committee states, in discussing the speed, "this may be checked by instantaneous readings or intermittent countings taken at regular intervals and numerically averaged." I think that clause should read *must be*. It is almost imperative in any test that we should have a definite check on the readings.

5 In Par. 60 attention is called to "how the cards shall be intergrated," as one of the important considerations. It is difficult to determine exactly the factors which influence the mean effective pressure of the cards. It is easy to have the indicator card so integrated that the brake horsepower will be larger than the indicated horsepower, and the work must be handled with much care.

6 In regard to the characteristics of coal, it might be of interest to have on file a brief outline made for the American Society of Testing Materials. That society intends to draw up specifications on the basis of this outline. It may also be wise to have on record the Test Code adopted by the Government for the plant of the Geological Survey, which though relating to a special plant may assist the committee to get at the general points which have proved of special importance in these investigations.

7 The writer is in sympathy with the committee's recommendation of the use of high actual heat value for the gas, but would like to inquire in regard to liquid fuel. In questions relating to alcohol and gasolene, particularly with alcohol in which the percentage of water is from 15 to 50 per cent, the efficiency curves, which are very beautiful, when worked out on the basis of low heat value, are completely destroyed, and present peculiar characteristics when referred to the high basis. Does the committee intend to include the liquid fuels?

A. A. CARY. I think it is entirely wrong to consider a producer gas plant and its engine as one continuous piece of apparatus, or practically so. In the days of Watt, the engine efficiency was determined by pounds of coal per indicated horse power and what happened between the coal pile and the steam engine indicator was entirely neglected. Finally engineers began to realize that there was a boiler between the coal pile and the engine indicator, which had a part in that efficiency, and we are now careful to get at the efficiency of both boiler and engine. The individual efficiency of both producer and

engine should be determined in producer gas practice, for only in that way can improvements be made.

EDWIN D. DREYFUS. In the re-arrangement of the report, from which in its final form such parts as have served merely to guide the committee in choosing and defining the most practical methods of testing will no doubt be eliminated, or incorporated in an appendix, a set of practical and standard tests should be distinctly defined, noting all refinements to be observed and the degree of accuracy to be attained.

2 The question of ratings (Par. 51) might be further elaborated. It is well known that altitude influences the power the engine is capable of developing, by reason of decreased density of mixture and the amount of heat entering the cylinder. Allowance for change in altitude should be specified, roughly about 3.3 per cent per 1000 ft. variation in elevation above sea level or point at which the normal rating is based, which is a fair average for fluctuating atmospheric conditions. Temperatures and pressures are additional factors. In some installations these items might be approximately selected, while in other cases conditions might arise to cause a considerable deviation from normal pressure and temperature. For example, if the supply pipes to the engines are located in a hot basement, and where the engine receives its supply through a holder located at an appreciable distance, maintaining a practically constant pressure, the drop in pressure in the delivery pipe becomes sufficient to cause the engine to pull several inches of suction at maximum load.

3 The question of reponsibility for the expense involved in conducting tests (Par. 87) is a commercial consideration and should not be included in the report. If, however, it is judged contrarily that this expense should be ascribed in some manner to protect the interest of the gas-power industry, the responsibility should then be differentiated in fairness to those builders who have already made large expenditures in proving the efficiency of apparatus, both in the field and in their shops. If small engines are concerned, witness tests may be readily made at the builders' works, which should involve no expense to the purchaser, provided they comply sufficiently with the requirements of the Standard Code of Tests. For large power work, the plant usually has an engineering staff capable of executing the tests as part of their duties, and in this case it would be optional to the builder to have representation when the tests were in progress, or at any time prior to the tests to provide normal adjustment of the apparatus.

4 Confusion in arriving at agreements (Par. 40*b*) and reporting the engine and producer tests separately would be likely to result from the abandonment of the use of the low value and of the distinction between the high and the low values of the gas. The low value has now been so extensively used in engine practice, that it should be clearly defined, and high and low values for different constituents of the gas also included. This will facilitate equating results from one basis to the other, where either one has been arbitrarily used in the past. This characteristic of the gas materially affects its suitability for the gas engine, due to the latent heat of the moisture formed in the burning of the hydrogen gases affecting the temperatures produced by the gas during combustion, and also a high hydrogen content limiting the compression pressures which can be safely used without danger of premature explosions. It should be given due regard, especially where the responsibility for the installation of the engine and producer is divided.

5 The percentage variation stipulated as permissible in the quality of the gas (Par. 40*d*) is more rigid than need be. The engine will work successfully on greater fluctuation, say from $7\frac{1}{2}$ to 10 per cent. Although a good producer will normally produce a quality of gas within these limits, somewhat greater variation might take place for short periods, which though causing no difficulty at the engine, may bring on a disagreement unless this condition is fully appreciated. Some attention should be given to defining distinctly how these measurements should be made.

L. B. Lent. Under general recommendations, Par. 77 reads: "All terms made in a guarantee should be defined. All guaranteed quantities must be capable of measurement and only one acceptable mode of measurement should be specified." Par. 79 reads: "Builders best serve their own interests when units are in terms most satisfactory to the purchaser, and involve only input and output for definite fuel, horsepower and time." If we were to put into a guarantee, which usually goes with the specifications of a gas engine, a description of all the apparatus used, stating the quantities and specifying the modes of measurement, our customers would be apt to entertain a proposition from some one with simpler specifications. If the Society sanctions a code in which the ordinary guarantee must be specified in legal phraseology, protecting the makers' interests until the apparatus is paid for, the guarantee would be far longer than the specifications and rather cumbersome. That part of the

report which deals with recommendations as to guarantees and other relations between manufacturers and builders, should be very carefully considered.

DR. C. E. LUCKE. This discussion will prove helpful to the Committee, and speaking for the Committee I can only hope there will be more of it. The more you consider these questions the more difficult they look. It is a simple matter to rise and recommend that a thing be done so and so, because you have not thought of any other way of doing it, but I suggest that you think about the report for six months, since it is only a preliminary report, and then after you have begun to realize the object of the report, write in your judgment of it.

2 The Committee has been nearly two years at this work and was many times seemingly ready to report, but did not because of the difficulty of the problems. The more views we can get on the questions at issue, the better. I have no intention of defending the report. My object is to ask you to give us your best judgment after mature deliberation, not before, because the questions are questions that are keeping courts busy all over this broad land today. The duty of the Committee was not to devise scientific tests for finding the physical actions involved—that is the business of the Code Committee. The business of this Committee was to eliminate misunderstanding between builders and purchasers, in so far as it could be traced to purely definitional causes from the use of terms having no accepted simple meaning, or from defining quantities not measurable.

3 I would therefore prefer not to reply to each of the suggestions but to let them go on record for the consideration of the Committee; but it is proper that I should say that there is the largest possible difference between tests for scientific information and the tests necessary to define a piece of apparatus sold. It is with the latter kind that this Committee deals and not with the former.

PRELIMINARY REPORT OF LITERATURE COMMITTEE

In accordance with a recommendation from the Chairman of the Meetings Committee of the Section, in a communication published in The Journal for October 1908, a Literature Committee was appointed at a meeting of the Gas Power Executive Committee, January 27, 1909. A tentative program for the work of the committee was outlined by Dr. C. H. Benjamin, *Chairman*, in a verbal report at the Spring Meeting of 1909. It is hoped to index all books, periodicals and transactions, domestic and foreign, dealing with gas power and allied subjects; also to present reviews of new books, and abstracts of important articles. Index cards have been prepared and are being filled by the members of the Committee, the work being divided under the following heads: reviews of French books and periodicals; reviews of German books and periodicals; reviews of Italian books and periodicals; revised list of articles appearing in English and American periodicals; reviews of English and American books as they appear; index of material on file in the library, both books and periodicals. The lists will be published from time to time in The Journal, with an index whenever the accumulation of material warrants.

GAS POWER LITERATURE

ARTICLES IN PERIODICALS[1]

ALCOOL, VINS E BIRRA, E. BOULLANGER. *L'Industria Rivista Tecnica, October 17, 1909.* 1½ pp. *bdfA*.

Deals with the commercial aspects of alcohol power.

AUTOMOBILE-MUFFLER FOR FREE EXHAUST. *Le Genie Civil, September 25, 1909.*

Note abstracting article in La Vie Automobile, July 17, 1909.

CALORIMETRY OF GASES, METHOD AND APPARATUS FOR. *Le Genie Civil, September 11, 1909.* ¼ p.

Abstract of note presented by M.M. Lemoult and Jungfleisch to the Academie des Sciences. A simplification based upon loss of volume and consumption of oxygen.

CALORIMETRY IN THE THERMAL LABORATORY OF THE UNIVERSITY OF MOSCOW, METHODS OF, LOUGUININE AND SCHÖNKAREW. *Journal de Chimie Phisique, October 1, 1909.*

Note reviewing the French translation of this Russian work, according to it very high authority. Trans. by Ter Gazarian, Paris: A. Hermann, 1908. 192 pp.

CHEMISTRY SINCE LAVOISIER, HISTORY OF DEVELOPMENT OF. *Revue Universelle des Mines et de la Metallurgy, August 1909.*

Note reviewing French translation of fourth German edition (the first appeared in 1869), by Professor Ladenburgh of the University of Breslau. *Paris*, 1909. P. Hermann.

COMBUSTIBILI. *L'Industria Rivista Tecnica, September 26, 1909.* 1 p. *abdfA*.

From Chemiker Zeitung, 1909, p. 893. Considers mineral oil and gas for illumination, and developments of the oil, gas and coke industry.

EXPLOSIONS, GASEOUS. *Engineering (London), September 3, 1909.* Report to British Association at Winnepeg. 4½ pp., 3 figs., 6 curves. *ceA*.

ENGINES, INFLUENCE OF COMPRESSION PRESSURE UPON THERMAL EFFICIENCIES OF GAS, W. A. TOOKEY. *Engineering (London), October 15, 1909.* 2 pp., 3 curves, 1 table. *cB*.

Discussion of report of Professor Burstall before Institution of Mechanical Engineers.

[1] Opinions expressed are those of the reviewer, not of the Society. Articles are classified as: *a* comparative; *b* descriptive; *c* experimental; *d* historical; *e* mathematical; *f* practical. A rating is occasionally given by the reviewer, as *A*, *B*, or *C*.

ENGINE, VARIABLE-STROKE PETROL. *The Engineer (London), October 15, 1909.* ¾ p., 3 figs., 1 table. *bcB.*

A new engine for motor cars giving variable speed on direct drive.

ENGINE, 600-B.H.P. "PREMIER" GAS. *Engineering (London), November 19, 1909.* ⅓ page, 1 fig., *bC.*

Three-cylinder, direct-connected, gas-engine; water-cooled piston, engine mechanically oiled, etc

ENGINES, LOW-TENSION IGNITION GEAR FOR GAS. *Engineering (London), October 29, 1909.* 1 p., 7 figs., *bC.*

Built for large engines; uses direct current from 65 to 220 volts. Intended to prevent possibility of pre-ignition.

ENGINE, NEW REVERSIBLE MARINE OIL. *The Engineer (London), December 10, 1909.* ¾ page, 3 figs., *bC.*

The "Bolinders" engine, by Jas. Pollock Sons & Co. Two-cylinder, two-cycle, hot-bulb, oil-injected, with reversing gear.

FUEL FOR AUTOMOBILES, BENZOL AS A CHEAP. *Le Genie Civil, October 2, 1909.*

Note abstracting tests reported in L'Automotor, June 26, 1909.

MOTORS, BALANCING AUTOMOBILE, *Le Genie Civil, September 25, 1909.*

Note abstracting article in La Technique Automobile, July 15, 1909.

MOTORS FOR DIRIGIBLES AND AEROPLANES, LIGHT EXPLOSION, Ch. Dantin *Le Genie Civil, June 5, 12, 19, 1909.* 18 pp., many figs. *bA.*

This article has not been inspected fully; but its great importance, as a résumé of French practice in this field, makes its inclusion inevitable.

MOTORS, SAFETY CRANK FOR HOISTS OR EXPLOSION. *Bulletin de la Société Industrielle de l'Est, July 1909.* 1 p.

Program for a competition for designs opened by the Association des Industriels de France contre les Accidents du Travail, 4 Blvd. St. André, Paris.

MOTORI A COMBUSTIONE INTERNA—MOTORI DIESEL REVERSIBILI. Ling R. Maraver. *L'Industria Rivista Tecnica, August 15, 1909.* 2 pp., 2 figs., *bcfA.*

Discussion of Diesel 3-cylinder gas engine.

MOTORI A COMBUSTIONE INTERNA. *L'Industria Rivista Tecnica, October 17 1909.* 2 cols., 1 fig., 1 table. *abcA.*

Shows complete sections of producer plant with engine of A. G. Dresdner Gas Motoren-Fabrik gia Moritz Hilledt Dresda.

MOTORE AD OLIO PESANTE. *L'Industria Rivista Tecnica, August 8, 1909.* 1½ pp., 2 figs., *bcfB.*

Discussion of gas engine of 3 to 15 h.p.

MOTORI A GAS E MOTORIA VAPORI. *L'Electricista, May 15, 1909.* 2 pp. *afB.*

Discussion of advantages of gas for power plant economy.

PUMP, AN INTERNAL-COMBUSTION, Herbert A. Humphrey. *Engineering (London), November 26, December 3, 10, 1909.* 13 pp. 17 figs., 13 curves. Also *The Engineer (London),* same dates. *bcfA.*

Paper before Institution of Mechanical Engineers. A new principle in pumping water and compressing air. The gas is exploded on the surface of the water in an explosion chamber in one end of a U-tube, the explosion forcing the water up in the other leg. No piston, connecting rod or crank, is used. Since the expansion stroke is longer than the compression, the gas can expand to atmospheric pressure.

PUMP, THE HUMPHREY GAS, W. Cawthorne Unwin. *Engineering (London), October 15, 1909.* 3½ pp. 8 figs., 4 tables. *bcA.*

Report of tests on Humphrey internal-combustion pump, with general conclusions.

BOOKS

ENGINE, THE GAS. F. R. Hutton. *Wiley, 1908.* 3d. ed. 562 pp., 5¾ x 9, 259 figs. $5.

A treatise on the theory, construction and operation of internal-combustion motors.

ENGINE, THE GAS. Forrest R. Jones. *Wiley, 1909.* 447 pp., 5¾ x 9¼, 142 figs., 30 tables. $4.

Construction, operation, theory and performance of gas engines. Particularly good are the parts dealing with operation, ignition, adjustment and the location of troubles. Useful and practical for the operator as well as for the student.

ENGINE, THE GAS. Cecil P. Poole. *Hill, 1909.* 97 pp., 6 x 9½, folding tables. $1.

Elementary theory, construction, functions of various parts and operation of gas engines. Practical, useful and concise. A good introductory work of strictly limited scope.

ENGINE IN PRINCIPLE AND PRACTICE, THE GAS. A. H. Goldingham. *Gas Power Publishing Company, 1907.* 195 pp., 6 x 9¼, 107 figs. $1.50.

A good elementary book for non-technical readers. No theory is included.

ENGINE THEORY AND DESIGN, GAS. A. C. Mehrtens. *Wiley, 1909.* 250 pp., 5 x 8¼, over 200 figs. $2.50.

Elementary and superficial book, not reliable in its fundamental parts. Contains descriptive matter, and working drawings of small two-cycle and four-cycle engines.

ENGINE, GAS, PETROL AND OIL, Dugald Clerk. *Wiley, 1909.* v., 1 VII + 380 pp., 8vo., 5 plates, 126 figs. *ce.* $4.

Historical sketch and thermodynamics of gas engines. A very complete study of the phenomena of gaseous explosions in closed vessels, with critical comparative examination of results. Followed by a study of explosion phenomena in actual engines by Clerk's "zig-zag" method—and the deduction therefrom of apparent specific heats of gases. Method developed for determining heat balances based on the "apparent" specific heat, and applied to various gas-engine tests. A valuable contribution to the theory of the gas engine—suitable for advanced students.

ENGINES, INTERNAL-COMBUSTION. R. C. Carpenter, H. Diederichs. *Van Nostrand, 1909.* 2d ed. 597 pp., 6 x 9½, numerous text figs. $5.

A comprehensive work of value, including the results of much German investigation for the first time in English. Four sections, dealing with: *a* theoretical considerations; *b* fuels and the phenomena of combustion; *c* construction and operation; *d* power estimation, tests and costs. A good text or reference book.

ENGINES, INTERNAL-COMBUSTION. Wm. H. Hogle. *McGraw, 1909.* 250 pp., 5½ x 8¾, 106 figs., 21 tables. $3.

Contains description and comparison of cyles; practical details of operation; starting devices; carburetors and vaporizers; producers; fuels and combustion; engine compression; the indicator card; design; governing devices; ignition; engine testing; report of tests. Treatment everywhere is elementary; rules of thumb only, given for design, and those only for small engines. Numerous inaccuracies.

ENGINES, CARBURETING AND COMBUSTION IN ALCOHOL. E. Sorel. *Wiley, 1907.* 269 pp., 5 x 8¼, 26 figs. *bcfA.* $3.

A valuable treatise on the properties of alcohol and on carbureting and explosions, based upon the author's extensive research. Unique in this field.

ENGINE, THE INTERNAL-COMBUSTION. H. E. Wimperis. *Van Nostrand, 1908.* 326 pp., 5½ x 8¾, 114 figs. $3.

Devoted primarily to the development of a new theory of gas engines, based upon variable specific heats of the gases. Much critical discussion of gaseous explosion experiments. Gas producers and engines are described rather briefly, and much information is given as to tests, costs, statistics and general data. Gasolene engines are also dealt with. The book takes up a number of special topics which are subjected to mathemetical analysis; as for example, the inertia effect in indicators, the theory of jet carburetors, etc. It is interesting and suggestive throughout and would be most valuable to advanced students.

MOTORS, OIL. G. Lieckfeld. *Lippincott, 1908.* 272 pp., 6 x 9¼, 306 figs. $4.50.

An English translation of a well-known German book. Deals with petroleum, benzol and alcohol engines. Properties of the fuels; historical development of the motors; construction of the parts; examples of modern construction, giving cost, weight, dimensions; applications for various purposes; erection and operation. The book is entirely descriptive, with no theory and but little explanation of functions of parts. A useful book in a limited field.

POWER, GAS. F. E. Junge. *Hill, 1908.* 548 pp., 6 x 9¼, 8 plates, 145 figs. *abdfA.* $5.

Part I: Evolution of gas power and economic aspects; includes a historical and critical study of the development of the present methods. Part II: Design and construction of large gas engines; with descriptions of the most prominent German types. Part III: Applications of gas power in iron and steel, coal-mining and coke-making industries, and elsewhere: use of low-grade fuels. A suggestive and useful treatment of the use of blast-furnace and coke-oven gases for developing large powers.

OTHER SOCIETIES

AMERICAN SOCIETY OF CIVIL ENGINEERS

At the meeting of the American Society of Civil Engineers on April 6, the New York tunnel extension of the Pennsylvania Railroad was discussed, with papers by B. F. Cresson, Jr., on The Terminal Station: West; and F. Lavis, on The Bergen Hill Tunnels. On April 20, Herbert M. Wilson presented a paper, entitled Federal Investigations of Mine Accidents, Structural Materials and Fuels, and the United States Testing Station at Pittsburg.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS

The 246th meeting of the American Institute of Electrical Engineers was held in Charlotte, N. C., March 30 to April 1. Papers were presented by A. Milmow, on Electric Drive in Textile Mills; by E. D. Latta, Jr., on Gas Engines in City Railway and Light Service; by A. E. Kennelly, on Modifications of Hering's Laws of Furnace Electrodes; by Carl Hering, on The Proportioning of Electrodes for Furnace Electrodes; by E. E. F. Creighton on Some Demonstrations of Lightning Phenomena; by W. S. Lee, on Economics of Hydroelectric Plants; by L. C. Nicholson, on Protecting Insulators from Lightning and Power Arc Effects on the Lines of the Niagara and Lockport Power Co. Visits were made to various plants in the vicinity, including a tour of the Great Falls and Rocky Creek Stations and a 100,000-volt sub-station of the Southern Power Company.

Through the Industrial Power Committee the Institute coöperated with The American Society of Mechanical Engineers in the meeting of April 12, a report of which is given on an earlier page.

At the regular April monthly meeting, which was postponed to April 15, Dr. Samuel Sheldon, professor of physics and electrical engineering at Brooklyn Polytechnic Institute, presented a paper, Education for Leadership in Electrical Engineering, which was discussed by prominent educators.

Circumstances have made it expedient to change the date of the proposed meeting at San Francisco to May 5–7. This meeting is to be under the auspices of the High-Tension Transmission Committee. Arrangements are being made for tours of inspection to important standby plants and receiving stations, transmission plants and mountain generating stations, during that and the following week. **The annual meeting of the Institute will be held in New York, May 17.**

On May 27, there will be a special meeting, with papers, under the auspices of the Railway Committee. A list of papers for these meetings will be found in another department.

The annual convention will be held June 27-30 at Jefferson, N. H.

AËRO CLUB OF AMERICA

Headquarters have been opened in the Engineering Societies Building by the Aëro Club of America, and the public was invited to the opening meeting in the Auditorium of the building, Wednesday, April 13. A short address by Cortlandt F. Bishop, president of the club, was followed by an exhibit of moving pictures showing recent leading events in aërial navigation and the possibilities of aërial warfare. Views of the club's new aviation field at Mineola, L. I., including the sheds in which members may house their aëroplanes and several machines now under construction, were also thrown on the screen. Following these exercises, a reception was held in the rooms of the club.

NATIONAL METAL TRADES ASSOCIATION

The twelfth annual convention of the National Metal Trades Association was held at the Hotel Astor, New York, April 13, 14, 1910. On Wednesday, besides the routine reports, William Lodge, chairman of the committee on Industrial Education, offered a report reviewing the work of the Winona Technical Institute at Indianapolis, Ind., to the support of which the Association contributes. The papers presented were: Cincinnati's Continuation School, by Dr. Frank B. Dyer; Employers' Liability Insurance, by Miles Dawson; The Growth of the Coöperating System, by Prof. Herman Schneider; Insurance against Unemployment, by John L. Griffiths. The papers presented at the Thursday morning session were: Modern Methods of Shop Management, by Fred. S. Waldron, Mem.Am.Soc.M.E.; Premium Systems, by Carl G. Barth, Mem.Am.Soc.M.E.; Cincinnati's Continuation School, by J. Howard Renshaw. The annual banquet took place Wednesday evening at the Hotel Astor. President Wells presided as toastmaster, and the diners numbered about two hundred and fifty. J. H. Schwacke was elected president of the association.

ENGINEERS SOCIETY OF WESTERN PENNSYLVANIA

Notice has been received at this office of the removal April 27 of the Engineers Society of Western Pennsylvania to their new quarters in the Sixth Avenue wing, 25th floor, of the Oliver Building, Sixth Avenue and Smithfield Street, Pittsburg, Pa.

PERSONALS

John R. Allen, professor of machanical engineering, University of Michigan, will head the engineering school to be established at Robert College, Constantinople. Professor Allen will go to Turkey early next summer.

A biographical sketch of Prof. L. P. Breckenridge was published in the April issue of *Cassier's Magazine.*

Edward W. Burgess, formerly in the employ of the Metzger Motor Car Company, Detroit, Mich., has become connected with the J. I. Threshing Machine Company, Racine, Wis.

Gordon M. Campbell, formerly superintendent of the power apparatus shops of the Western Electric Co., Chicago, Ill., has become connected with the turbine department of the General Electric Co., West Lynn, Mass.

A. W. Cash, mechanical engineer and expert, recently located at Newark, N. J., has taken charge of the regulating valve and engineering department of the H. Mueller Mfg. Co., Decatur, Ill.

Walter Castanedo has been appointed manager for the New Orleans, La., district of the Harrisburg Foundry and Machine Works. He was formerly a member of the firm of Glenny & Castanedo, New Orleans, La.

S. M. Chandler has been appointed president of the Chandler-Boyd Supply Company, Pittsburg, Pa. He was formerly associated with the Pittsburg Valve and Fittings Company, Barberton, O.

Philip G. Darling, recently mechanical engineer of Manning, Maxwell & Moore, Inc., New York, has become associated with the development department of the E. I. du Pont de Nemours Powder Co., Wilmington, Del.

William W. Estes has accepted a position with the General Fire Extinguisher Company, Providence, R. I., as designer. Mr. Estes was until recently chief engineer of the R. I. Co., of the same city.

A. L. G. Fritz has become chief draftsman of the Hartford Suspension Company, Jersey City, N. J. Mr. Fritz was formerly associated with the Tee Square and Triangle Co., Newark, N. J.

Reuben Hill, formerly factory manager of the Bristol Engineering Corporation, Bristol, Conn., has entered the service of the Hudson Motor Car Company, Detroit, Mich.

Dr. D. S. Jacobus read a paper on Superheated Steam and Superheaters at the annual meeting of the National Association of Cotton Manufacturers, Boston, April 27, 28.

Charles Kirchhoff sailed for Europe April 19, expecting to be abroad for several months. He will sojourn for some time in Italy, and from there go to Germany.

Charles W. Lummis, recently mechanical engineer of the Camden Iron Works, Camden, N. J., has become identified with the Scovill Manufacturing Co., Waterbury, Conn.

Walter M. McFarland, who has occupied the office of acting vice-president for the Westinghouse Electric & Mfg. Co., Pittsburg, Pa., for a period extending over ten years, has resigned to accept an official position with the Babcock & Wilcox Co., Singer Building, New York.

D. H. Macdonald has entered the service of the Southern Engine and Boiler Works, Jackson, Tenn. He was formerly identified with the Minneapolis Steel and Machinery Company, Minneapolis, Minn., as general superintendent of the mechanical department.

John B. Mayo, formerly connected with the Crocker-Wheeler Co., Ampere, N. J., has been appointed mechanical engineer and chief draftsman of the Texas Portland Cement Company, Cement, Texas.

Fred J. Miller has been elected a member of the Board of Directors of the Union Typewriter Company.

Albert B. Moore, formerly chief engineer of the Griffin Wheel Company, Chicago, Ill., has resigned his position and formed a partnership with Andrew W. Woodman, under the firm name of Woodman & Moore, civil and mechanical engineers, with offices in the People's Gas Building, Chicago, Ill.

Francis P. Ritchie has accepted the position of factory manager of the Berliner Gramophone Company, Montreal, Canada. He was recently superintendent of the Northern Electric and Mfg. Co., of the same city.

William J. Sando, manager of the pumping engine and hydraulic turbine department of the Allis-Chalmers Co., for nearly six years, has resigned. After taking a few months' rest Mr. Sando will open an office in Boston as consulting engineer.

Walter G. Scott, recently associated with the Allis-Chalmers Co., West Allis, Wis., has become identified with the Cyclone Drill Company, Orville, O.

John J. Swan has become connected with the Keller Mfg. Co., of Philadelphia, Pa. Until recently he was secretary of the W. P. Pressinger Co., New York.

Charles E. Sweet, formerly superintendent of steam turbine construction, Westinghouse Machine Company, East Pittsburg, Pa., has become associated with the E. M. F. Co., Detroit, Mich.

Charles W. Werst has accepted a position with the Lima Locomotive and Machine Co., Lima, O., in the capacity of assistant superintendent. He was until recently general foreman of the Baldwin Locomotive Works, Philadelphia, Pa.

CURRENT BOOKS

AMERICAN MACHINISTS' HANDBOOK AND DICTIONARY OF SHOP TERMS. A Reference Book of Machine Shop and Drawing Room Data, Methods and Definitions. By Fred H. Colvin and Frank A. Stanley. *New York: McGraw-Hill Book Co., 1909.* Morocco, pocket-book size, xxi +513 pp., illustrated. Price, $3.

Contents: Screw Threads: Cutting Screw Threads, Standard Proportions of Screw Threads, Measuring Screw Threads; Pipe and Pipe Threads; Twist Drills and Taps; Taps; Files; Work Benches; Soldering; Gearing; Milling and Milling Cutters; Cam Milling Machine Feeds and Speeds, Tables for Use with the Dividing Head, Milling Cutter, Reamer and Tap Flutes; Grinding and Lapping; Grinding Wheels and Grinding, Lapping, Reamer and Cutter Grinding; Screw Machine Tools, Speeds and Feeds: Types of Tools and their Construction, Speeds and Feeds for Screw Machine Work; Punch Press Tools; Bolts, Nuts and Screws: Tables of Cap and Machine Screw Dimensions, Tables of A. S. M.E. Standard Machine Screw Dimensions, Nut and Bolt Tables, Miscellaneous Tables; Calipering and Fitting: Press and Running Fits, Dimensions of Keys and Key-Seats; Tapers and Dovetails: Measuring Tapers, Tables of Standard Tapers; Shop and Drawing Room Standards: Standard Jig Parts, Tables of Dimensions of Standard Machine Parts, Miscellaneous Tables; Wire Gages and Stock Weights; Belts and Shafting; Steel and Other Metals; General Reference Tables; Shop Trigonometry; Dictionary of Shop Terms.

THE AMERICAN PRACTICE OF GAS PIPING AND GAS LIGHTING IN BUILDINGS. By Wm. Paul Gebhard. *New York, McGraw-Hill Book Co., 1908.* Cloth, 8vo, 306 pp. Price, $3.

Contents: Prejudices against the Use of Gas; Popular Fallacies about Gas; Advantages of Gas as an Illuminant; Advantages of Gas as a Source of Heat and Power; The Arrangement of Gas Piping in Buildings; Specification for Gas Piping for Coal or Water Gas; Rules, Tables and Regulations of Gas Companies and of Building Departments; Piping for Natural Gas; Piping for Air Gas or Gasolene Machine Gas; Piping for Acetylene Gas; The Testing of Gas Pipes; Gas-Light Illumination; Gas Burners; Gas-Pressure Regulation; Gas Globes and Globe Holders; Gas Fixtures; Gas Meters and Gas Meter Stories; The Illumination of Interiors with Gas Lights; The Lighting of Country Houses; The Relations between Gas Companies and Gas Consumers; Practical Hints for Gas Consumers; Some Facts about the Gas Supply; Accidents with Gas; Dangers to the Public Health from Illuminating and Fuel Gas; Historical Notes on the Development and Progress of the Gas Industry; Bibliography of Gas Lighting.

THE DESIGN AND CONSTRUCTION OF OIL ENGINES, with full directions for erecting, testing, installing, running and repairing; including descriptions of American and English kerosene oil engines. By A. H. Goldingham. 3d edition, revised and enlarged. *New York, Spon & Chamberlain, 1910.* Cloth, 12mo, viii+260 pp., illustrated. Price, $2.50.

Contents: Introductory; On Designing Oil Engines; Testing Engines; Cooling Water Tanks and Other Details; Oil Engines Driving Dynamos; Oil Engines Connected to Air Compressors, Water-Pumps, etc.; Instructions for Running Oil Engines; Repairs; Oil Engine Troubles; Various Engines Described; Portable Engines; Large Sized Engines; Fuels; Miscellaneous.

PRACTICE AND THEORY OF THE INJECTOR. By Strickland L. Kneass. 3d edition revised and enlarged. *New York, John Wiley & Sons, 1910.* Cloth, 8vo, 175 pp., illustrated. Price, $1.50.

Contents: Early History; Development of the Principle; Definition of Terms-Description of the Important Parts of the Injector; The Delivery Tube; The Combining Tube; The Steam Nozzle; The Action of the Injector; Application of the Injector—American and Foreign Practice; Determination of Size—Tests; Requirements of Modern Railroad Practice—Repairs—Methods of Feeding Locomotive Boilers; Feed Water Heating—Efficient Feeding—Flue Mileage—Scale-Bearing Water—Check Valves.

PUMPING ENGINES FOR WATER WORKS. By Charles Arthur Hague. *New York, McGraw-Hill Book Co., 1907.* Cloth, 8vo, xi + 372 pp., illustrated. Price, $5.

Contents: The Pumping Engine; Historical; Economic Steam Duty; The Advent of Triple Expansion; The Mariotte Curve; Steam Jackets; Coal Duty of Pumping Engines; Actual Conditions of Pumping; The Worthington Duplex Pumping Engine; The Gaskill Pumping Engine; The Reynolds Triple Expansion Pumping Engine; Various Types and Classes; Pumping Engines adapted to Conditions; Installation of Pumping Engines; Investment Value of Pumping Engines; Suction Lift and Suction Pipes; Water Passages and Water Valves; The Water Plungers; Air Chambers; Steam Piston; Steam Cylinders; Cross Heads; Frames and Bedplates; Material for Pumping Engines; Duty Tests of Pumping Engines.

DRAWINGS FOR MEDIUM SIZED REPETITION WORK. With Examples for Motor Car parts. By R. D. Spinney *London, E. & F. N. Spon, Ltd., 1909.* Cloth 8 vo, + 91 pp., illustrated. Price. $1.50.

Contents: Drawings Generally; Standard Lists; Indexes; Tolerances; Dimensioning; Notes on Designing for Repetition Work; Drawing Office Routine.

ACCESSIONS TO THE LIBRARY

This list includes only accessions to the library of this Society, included in the Engineering Library. Lists of accessions to the libraries of the A.I.E.E. and A.I.M.E. can be secured on request from Calvin W. Rice, Secretary, Am.Soc.M.E.

AMERICAN MACHINISTS' HANDBOOK AND DICTIONARY OF SHOP TERMS. By F. H. Colvin and F. A. Stanley. *New York, McGraw-Hill Book Company, 1909.*

AMERICAN PRACTICE OF GAS PIPING AND GAS LIGHTING IN BUILDINGS. By W. P. Gerhard. *New York, McGraw-Hill Book Company, 1908.*

AMERICAN RAILWAY ASSOCIATION. Statistical bulletins Nos. 66, 67-B. *Chicago, 1910.*

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS. The Journal, vol. 31, Nos. 1-12. 2 vols. *New York, 1909.*

BALLISTIC ELECTRO-DYNAMOMETER METHOD OF MEASURING HYSTERESIS LOSS IN IRON. University of Kansas, Engineering Experiment Station, Bulletin No. 1. By M. E. Rice and Burton McCollum. *Lawrence, 1909.* (Gift.)

BRIQUETTED COAL AND ITS VALUE AS A RAILROAD FUEL By C. T. Malcolmson.

DESIGN AND CONSTRUCTION OF OIL ENGINES. Ed. 3. By A. H. Goldingham. *New York, Spon & Chamberlain, 1910.*

DESIGN OF TURBO FIELD MAGNETS FOR ALTERNATE-CURRENT GENERATORS, WITH SPECIAL REFERENCE TO LARGE UNITS AT HIGH SPEEDS. By Miles Walker. Institution of Electrical Engineers, 1909. (Gift of Calvin W. Rice.)

DETROIT BOARD OF WATER COMMISSIONERS. 57th annual report. *Detroit, 1909.* (Gift.)

DRAWINGS FOR MEDIUM-SIZED REPETITION WORK. By R. D. Spinney. *New York, Spon & Chamberlain, 1909.*

ELECTRIC RAILWAY AND LIGHTING PROPERTIES. By Stone & Webster. *Boston, 1910.* (Gift of Calvin W. Rice.)

EPITOME OF THE WORK OF THE AERONAUTIC SOCIETY FROM JULY 1908 TO DECEMBER 1909. (Bulletin no. 1) *New York, 1909.* (Gift of Aeronautic Society, New York.)

FIFTH AVENUE BUILDING SITE OF FIFTH AVE. HOTEL, MADISON SQUARE, NEW YORK. *New York.* (Gift of Fifth Avenue Building Company.)

GEOLOGY OF THE AUBURN-GENOA QUADRANGLES. (New York State Museum, Bulletin no. 137.) By D. D. Luther. *Albany, 1910.* (Gift.)

HAWAII. Government and conditions before and since annexation to the United States and present requirements. Speech of Chauncey M. Depew, February 24, 1910. *Washington, 1910.* (Gift).

INNOKO GOLD-PLACER DISTRICT, ALASKA. (Bulletin no. 410, U. S. Geological Survey.) By A. G. Maddren. *Washington, 1910.* (Gift.)

IRON ORES, FUELS AND FLUXES OF THE BIRMINGHAM DISTRICT, ALABAMA (Bulletin no. 400, U. S. Geological Survey.) By E. F. Burchard and Charles Butts. *Washington, 1910.* (Gift.)

LEGAL AID SOCIETY. 24th annual report of the president, treasurer and attorneys. *New York, 1909.* (Gift.)

LOCOMOTIVE COALING STATIONS. (Bulletin no. 15, Roberts & Schaefer Co., Chicago.) *Chicago.* (Gift of author.)

MODERN COAL MINING PLANTS AND WASHERIES. (Bulletin no. 17, Roberts & Schaefer Co., Chicago.) *Chicago.* (Gift of author.)

PALEONTOLOGY OF THE COALINGA DISTRICT, FRESNO AND KINGS COUNTIES, CALIFORNIA. (Bulletin no. 396, U. S. Geological Survey.) By Ralph Arnold. *Washington, 1909.* (Gift.)

PLANNING AND BUILDING OF INDUSTRIAL PLANTS. Section II. By Charles Day. *Dodge & Day Philadelphia, 1909.*
TYPICAL OPERATIONS. No. 2.

POSSIBILITIES OF ELECTRICAL POWER TRANSMISSION FOR MAIN PROPULSIONS AND SPEED REGULATION. By W. P. Durtnall. (Gift of author.)

PRACTICE AND THEORY OF THE INJECTOR. Ed. 3. By S. L. Kneass. *New York, J. Wiley & Sons, 1910.*

PROGRESS OF ELECTRICAL BRAKING ON THE GLASGOW CORPORATION TRAMWAY SYSTEM. Institution of Electrical Engineers, 1910. By A. Gerrard. (Gift of Calvin W. Rice.)

RECONNAISSANCE OF SOME MINING CAMPS IN ELKO, LANDER AND EUREKA COUNTIES, NEVADA. (Bulletin no. 408, U. S. Geological Survey.) By W. H. Emmons. *Washington, 1910.* (Gift.)

RESULTS OF SPIRIT-LEVELING IN ILLINOIS, 1896 TO 1908, INCLUSIVE. (Bulletin no. 421, U. S. Geological Survey.) *Washington, 1910.* (Gift.)

SHORT-CIRCUITING OF LARGE ELECTRIC GENERATORS AND THE RESULTING FORCES ON ARMATURE WINDINGS. By Miles Walker. Institution of Electrical Engineers, 1909. (Gift of Calvin W. Rice.)

TABLES OF EFFICIENCIES IN PERCENTAGES, BUTT AND DOUBLE STRAP JOINTS. Computed by W. W. Ramsay. *Boston 1910.* (Gift of Commonwealth of Massachusetts Board of Boiler Rules.)

TESTS OF VERTICAL PUMPING AT QUACKENBUSH PUMPING STATION, ALBANY N. Y. *Chicago, 1909.* (Gift of H. A. Allen & Co.)

TRINITY'S TENEMENTS. CONDENSED REPORT. *New York, 1909.* (Gift of Corporation of Trinity Church.)

EXCHANGES

AMERICAN GAS INSTITUTE. Directory of Membership, 1909. *Easton, 1909.*

AMERICAN SOCIETY OF CIVIL ENGINEERS. Constitution and List of Members, February 1910. *New York, 1910.*

AMERICAN SOCIETY OF CIVIL ENGINEERS. Transactions, vol. 66. *New York, 1910.*

AMERICAN STREET AND INTERURBAN RAILWAY ASSOCIATION. Membership list, 1910. *New York, 1910.*

INSTITUTION OF CIVIL ENGINEERS. Minutes of proceedings, vol. 179. *London, 1910.*

INSTITUTION OF ENGINEERS AND SHIPBUILDERS IN SCOTLAND. Transactions vol. 52. *Glasgow, 1909.*

MACHINERY MARKET. March 4, 1910-date *London, 1910-date.*

NATIONAL ASSOCIATION OF COTTON MANUFACTURERS. Transactions, no. 87. *Boston, 1910.*

NEW YORK STATE MUSEUM. 62d annual report, vol. 1, 1908. *Albany, 1909.*

ROYAL SOCIETY OF NEW SOUTH WALES. Journal, vol. 42, 43, pt. 1. *Sydney, 1908-1909.*

THERMAL CONDUCTIVITY OF FIRE-CLAY AT HIGH TEMPERATURES. University of Illinois, Engineering Experiment Station, Bulletin no. 36. By J. K. Clement and W. L. Egy. *Urbana, 1909.*

UNITED ENGINEERING SOCIETY

INTERNATIONAL WHO'S WHO, 1910-1911. *New York, International Who's Who Publishing Company, 1910.*

DESIGNING AND DETAILING OF SIMPLE STEEL STRUCTURES. By C. T. Morris. *Columbus, 1909.*

GIFT OF SCHNEIDER & CO.

Acier au Manganese, propriétés et applications. Extract from Le Génie Civil. *Paris, 1908.*

Acier au Manganese. *1909.*

Acier Speciaux pour Pièces d'Automobiles. Objets exposés. *Paris, 1906.*

Album des Fers et Aciers. *1895.*

Etablissements de MM. Schneider & Cie. *Nevers, 1900.*
Materiel automobile. *1909.*
Materiel Electrique à Courants Alternatifs. Alternateurs. *1909.*
——— Dynamos Schneider Type "*E*" 1909, type "*S*" 1909.
——— Moteurs triphases, 1905.
——— Transformateurs, 1907.

PRINCIPALES INSTALLATIONS COMPORTANT DES DYNAMOS "SCHNEIDER" A COURANT CONTINU. Liste des references, 1905.

ANNEXE NO 1. *1906-1907.*

PRINCIPALES INSTALLATIONS COMPORTANT DU MATERIAL ELECTRIQUE À COURANTS ALTERNATIFS DE NOTRE DERNIER TYPE. Liste des references, 1906.

ANNEXE NO. 1. *1907.*

TRAVAUX D'AMELIORATION DU PORT DU HAVRE. *Paris, 1907.*

TRADE CATALOGUES

AVERY SCALE COMPANY, *North Milwaukee, Wis.* Automatic coal weighing, 24 pp.

BLAKE & KNOWLES STEAM PUMP WORKS, *New York.* The Blake-Knowles open feed water heaters, 12 pp.

COMPAGNIE DES FORGES ET ACIÉRIES DE LA MARINE ET D'HOMECOURT. *France.* Iron and steel parts for automobiles for buildings, castings of all kinds, wheels, projectiles, guns, fire arms. *75pp.*

G. DROUVÉ CO., *Bridgeport, Conn.* The anti-pluvius puttyless skylight, and the Lovell window operator, 6 pp.

GENERAL ELECTRIC COMPANY, *Schenectady, N. Y.* Price list No. 5214: G. E. Edison, carbon incandescent lamps, 7 pp; No. 5218: G. E. train-lighting lamps, 3pp.; Mazda vs. Tungsten lamps, 6 pp.; Bulletin No. 4716: Thomson watt-hour meters with prepayment attachments for direct and alternating currents, 7 pp.; No. 4719, G.E. fan motors and small power motors, 36 pp.

GOLDSCHMIDT-THERMIT COMPANY, *New York.* Reactions—first quarter, a quarterly periodical devoted to the science of aluminothermics, 20 pp.

GREENE, KETCHUM & CO., *New York.* Sullivan smokeless furnace equipment, 16 pp.

H. G. HAMMETT, *Troy, N. Y.* Trojan metallic packing for locomotive piston rods and valve-stems, 8 pp.; Richardson locomotive valves, 15 pp.; Trojan bell ringer for locomotives, 2 pp.; Samson bell ringer for locomotives, 2 pp.; Link grinders, 2 pp.; Radias grinder, 2 pp.; Triple-valve bushing roller, 2 pp.

IRONWORKS COMPANY, *Jersey City, N. J.* Crowe mechanical stokers for stationary, marine or locomotive boilers, 36 pp.

ANDREW J. MORSE & SON, *Boston, Mass.* Illustrated catalogue of diving apparatus and other submarine appliances, 72 pp.

NASON MFG. CO., *New York.* Nason "Vesuvius" steam trap, 8 pp.; Class B, Class C, and sidelug steam traps, 7 pp.

NATIONAL LOCK WASHER COMPANY, *Newark, N. J.* Catalogue of car curtains, curtain fixtures, sash locks, sash balances, nut locks, etc., 40 pp.

NAZEL ENGINEERING AND MACHINE WORKS, *Philadelphia, Pa.* Béché patent pneumatic power hammer, 12 pp.

NEW JERSEY CAR SPRING AND RUBBER COMPANY, *Jersey City, N. J.* Rubber goods, as fire hose, packing, belts, mats, gaskets, etc., 132 pp.

NEW YORK CONTINENTAL JEWEL FILTRATION CO., *New York.* Mechanical filtration, water purification, filtration of public water supplies, 64 pp.

NORTH WESTERN EXPANDED METAL CO., *Chicago, Ill.* Expanded metal in use on highway bridges, 16 pp.; Designing data for the use of expanded metal and expanded metal lath, 70 pp.

REAGAN GRATE BAR CO., *Philadelphia, Pa.* Improved chopping grates for stationary boilers, 40 pp.; Improved chopping grates for marine and locomotive boilers, 28 pp.

ROBERTS FILTER MFG. CO., *Philadelphia, Pa.* Pressure water filters and filter appurtenances, 72 pp.; Improved water filter for domestic use, 12 pp.

ROBERTS & SCHAEFER CO., *Chicago.* Bulletin No. 17: Construction of coal-mining plants, coal-storage plants, mine-ventilating plants, in U. S., 116 pp.; Bulletin No. 15: Construction work of the railroad department, during 1905–1907, 75 pp.

TEMPLETON MFG. CO., *Boston, Mass.* Sterling steam trap, 7 pp.

UNDER-FEED STOKER COMPANY OF AMERICA, *Chicago.* Publicity Magazine, March 1910, devoted to the interests of the Jones stoker, 16 pp.

EMPLOYMENT BULLETIN

The Society has always considered it a special obligation and pleasant duty to be the medium of securing better positions for its members. The Secretary gives this his personal attention and is most anxious to receive requests both for positions and for men available. Notices are not repeated except upon special request. Copy for notices in this Bulletin should be received before the 15th of the month. The list of men available is made up of members of the Society and these are on file, with the names of other good men not members of the Society, who are capable of filling responsible positions. Information will be sent upon application.

POSITIONS AVAILABLE

022 Engineer experienced in the design and construction of single and multistage centrifugal and turbo-pumps. Must be thoroughly competent to make estimates and prepare complete calculations and data for drafting room. Location, Middle West.

023 Chief draftsman on hydraulic-turbine machinery and hydro-electric plants; one capable of handling draftsmen and pushing work through the drafting room in a systematic and economical manner. Location, Middle West.

024 Instructor in mechanical and architectural drawing, Tuesday and Thursday evenings, October to May. Drafting room practice and some experience in teaching necessary. Location, one hour from Manhattan.

025 Glass company, Western Pennsylvania, requires technical man for general engineering work covering testing of furnaces, boilers and producers; advice on purchase of materials and supplies; design and supervision of new construction. A good working knowledge of chemistry and several years experience in similar work desirable.

MEN AVAILABLE

58 Member, Cornell graduate; twelve years varied experience electrical construction, steam and electrical power-plant equipment, operation and repairs, ice-making and refrigeration engineering and construction; desires position as chief engineer or master mechanic of large industrial or ice and cold storage plant.

59 Member, fifteen years experience in design and construction; seven years teaching; now professor of Mechanical Engineering in the South, wants similar position in the North or Central West or with an established engineering concern.

60 Member, graduate in mechanical engineering with degrees of B.S. and M.E.; two years practical experience in machine design and construction; five years as instructor in mechanical engineering in State university; desires position as assistant professor of mechanical engineering or of machine design. Location immaterial.

61 Member of the Society, graduate engineer, age 57, fifteen years practical experience, drafting room, foundry and machine shop; selling machinery; desires position as general superintendent or manager, where interest in profitable business can be acquired. At present employed.

62 Associate member, age 33, five years practical experience in manufacturing, now head of mechanical engineering department of a Southern technical college, desires position for the summer months with some good manufacturing firm. Would prefer to act as representative in some part of the South, but will consider work of other kinds. Specially qualified along the lines of gas and steam engines and of refrigerating machinery.

63 Member, wants position as general superintendent or works manager; age 35, practical mechanic, technical education, good hustler and organizer; now superintendent of plant doing heaviest class of iron and steel work. At liberty June first, $5000 a year.

64 Junior, technical graduate, several years experience in shop, office, drafting room and testing, with unusual executive ability, force and adaptability, desires to make change.

65 Associate, technical graduate, two years mechanical and electrical draftsman, five years experience in teaching and engineering research, at present engaged in teaching, desires a change. Would like position as head of department of mechanical engineering in some small technical school offering possibilities of growth and development, or as professor or assistant professor of experimental engineering in some large college desiring to develop laboratory and research work. Salary expected $2200.

66 Responsible position in a prominent university desired by engineer; for the past nine years head of a college of Mechanical and Electrical Engineering. Good reasons for changing, record open for inspection.

67 Member, engineer graduate of U. S. Naval Academy, large and influential engineering acquaintance and broad experience in correspondence, selling, and manufacturing in large and well-known works, holding executive positions of responsibility, desires a position of trust with a good manufacturing concern, or responsible position on the commercial end of a business, as branch manager in Boston or elsewhere.

68 Member, chief draftsman in or near New York City. Experience in steam pumps, air compressors, condensers, Corliss engines.

69 Member, graduate M.E. and C.E., twenty-three years experience in general engineering, ten in hydroelectric developments. Charge of the design and installation of power plants in this and other countries; desires position, East preferred.

70 Member desires to communicate with concerns preferably in Pennsylvania, established in general powerplant engineering and contracting work, with a view of buying an interest and assuming an active part in the business; or taking an interest in some manufacturing concern building power-plant equipment or accessories.

71 Technical graduate, experienced in several varied lines of industry, holding executive positions of responsibility during the last eight or nine years, desires to become associated in position of trust with good manufacturing concern, preferably located in the East or Middle West. Best of references.

72 Sales manager, general manager or assistant with manufacturer of power plant apparatus, "water-tube" boiler manufacturer preferred; Junior member, age 32. References can be furnished that will be satisfactory to the most critical. Preferred location, Philadelphia or New York.

CHANGES IN MEMBERSHIP

CHANGES OF ADDRESS

BOCORSELSKI, F. E. (1907), Asst. Mech. Supt., Am. Loco. Co., and 1843 W. Grace St., Richmond, Va.

BRYAN, William H. (1891), Cons. Mech. and Elec. Engr., 315 and 316 Title Guaranty Bldg., and 5718 Vernon Ave., St. Louis, Mo.

BURGESS, Edward W. (Junior, 1908), J. I. Case Threshing Mch. Co., Racine, Wis.

BURGOON, Charles Eli (1907), Ch. Engr., U. S. Court House and P. O. Bldg., and 3824 Rokeby St., Chicago, Ill.

CARLE, Nathaniel A. (1907), Cons. and Contr. Engr., 510 Central Bldg., Seattle, Wash.

CARPENTER, Allan O. (Associate, 1909), Granville Center, Bradford Co., Pa.

CASTANEDO, Walter (1907; 1909), Dist. Mgr., Harrisburg Fdy. & Mch. Wks., 1103 Hennen Bldg., and *for mail*, 1514 Peters Ave., New Orleans, La.

CHANDLER, Sellers McKee (Junior, 1905), Pres., Chandler-Boyd Supply Co., Terminal Warehouses, and *for mail*, 741 Browne St., Shadyside, Pittsburg, Pa.

DARLING, Philip G. (Associate, 1906), Development Dept., E. I. du Pont de Nemours Powder Co., Wilmington, Del.

ENGLISH, Harry K. (Associate, 1908), Genl. Elec. Co., Monadnock Bldg., Chicago, Ill.

FRITZ, Aime L. G. (Junior, 1907), Ch. Draftsman, Hartford Suspension Co., 150 Bay St., Jersey City, N. J.

GERRISH, William H. (1901), 15 Gould Ave., Malden, Mass.

GORDON, Frederic W. (1880), Clardon, Orchard Lane, Ft. Washington, Pa.

GUMP, Walter B. (Junior, 1902), Mech. and Elec. Engr., 206 Kerckhoff Bldg., Los Angeles, Cal.

HAGY, J. L. (Junior, 1901), Harrison Safety Boiler Wks., Tioga, and *for mail*, 1525 Montgomery Ave., Philadelphia, Pa.

HENES, Harry Wm. (Junior, 1909), 557 Barry Ave., Chicago, Ill.

HENSHAW, Frederick V. (1900), Cons. Engr., 24 Broad St., New York, and 79 State St., Brooklyn, N. Y.

HILL, Reuben (1908), Hudson Motor Car Co., Detroit, Mich.

JONES, Charles E. (1894), Mech. Engr., Instr. in Forging, Manual Training Sch. of Washington Univ., 5361 Von Verson Ave., St. Louis, Mo.

KEAN, A. J. A. (1908), Ch. Operating Engr., Guanajuato Power & Elec. Co., Apartado 50, Guanajuato, Mexico.

KENYON, Alfred Lewis (1904), care Dr. A. C. Griggs, Warrensburg, Mo.

KREUTZBERG, Otto August (Associate, 1904), V. P., Pfannmueller Engrg. Co., 3701 S. Ashland Ave., and 38 Roslyn Pl., Chicago, Ill.

LAMBIE, James M. (Junior, 1908), Asst. Genl. Mgr., Findlay Clay Pot Co., Findlay, O., and *for mail*, 306 E. Maiden St., Washington, Pa.

LUCAS, Henry M. (1904), Lucas Mch. Tool Co., E. 99th St. and L. S. & M. S. R. R., Cleveland, and *for mail*, 82 Page Ave., East Cleveland, O.

LUMMIS, Charles W. (Associate, 1907), Scovill Mfg. Co., Waterbury, Conn.

MACDONALD, Duncan H. (1903), Southern Eng. & Boiler Wks., Jackson, Tenn.

MASON, William (1880), 78 Burncoat St., Worcester, Mass.

MAYO, John B. (1892;1894), Mech. Engr. and Ch. Draftsman, Texas Portland Cement Co., Cement, Tex.

MERRITT, Joseph (1907), Cons. and Constr. Mech. Engr., 60 Prospect St., and 64 Deerfield Ave., Hartford, Conn.

MOORE, Albert B. (1903), Member of Firm, Woodman & Moore, Civ. and Mech. Engrs., Peoples Gas Bldg., and *for mail*, 5426 Jefferson Ave., Chicago, Ill.

NORRIS, William H., Jr. (Junior, 1909), Engr., W. R. Grace & Co., Lima, Peru.

OLIVER, E. C. (Junior, 1902), 475 Second Ave., Detroit, Mich.

OWEN, Ira June (Junior, 1905), Cons. Engr., Rm. 502, 115 Dearborn St., Chicago, and 110 Maple Ave., Oak Park, Ill.

PARK, Walter E. (1903), Young & Park, 45 Broadway, New York, N. Y.

PITKIN, Joseph Lovell (Associate, 1903), 72 Washington St., Atlanta, Ga.

PRESSINGER, W. P. (Associate, 1903), V. P. and Mgr. of Sales, Keller Mfg. Co., 21st St. and Allegheny Ave., Philadelphia, Pa.

RIGDON, Carl (Junior, 1907), 710 Market St., Chattanooga, Tenn.

RITCHIE, Francis P. (Associate, 1908), Factory Mgr., Berliner Gramophone Co., and *for mail*, 32 Sussex Ave., Montreal, Canada.

ROGERS, Charles Edward (Associate, 1898), Genl. Mgr., Fraser & Chalmers, Ltd., Corner House, and The Pines, Gordon Hill Rd., Parktown, Johannesburg, South Africa.

ROGERS, Robert W. (Junior, 1908), Mech. Engr., C. A. Stickney Co., and *for mail*, The Meadows, Dodd Pl., St. Paul, Minn.

RYDER, Malcolm P. (Associate, 1901), Engr., Witherbee Igniter Co., Springfield, Mass., and 11 Warren St., White Plains, N. Y.

SHALLENBERGER, Louis R. (1895; 1902), Box 55, Fenton, Mich.

STIMSON, Oscar M. (1906), O. M. Stimson & Co., First Natl. Bank Bldg., Chicago, Ill.

STREET, Clement F. (1893), with firm of Clement F. Street, Locomotive Stokers, 1427 Schofield Bldg., Cleveland, O.

SWAN, John Joseph (1899; 1909), Keller Mfg. Co., 21st and Lippincott Sts., Philadelphia, Pa., and Plainfield, N. J.

SWEET, Charles E. (1907), E. M. F. Co., Detroit, Mich.

SWENSON, Bernard Victor (1903), Barron G. Collier, Inc., 175 Fifth Ave., New York, N. Y.

THOMAS, Fred H. (Junior, 1909), Sales Engr., C. & G. Cooper Co., Mt. Vernon, O.

WALKER, Frank A. (Junior, 1909), with B. B. & R. Knight, and *for mail*, 46 Ring St., Providence, R. I.

WERST, Chas. Wm. (1909), Asst. Supt., Lima Loco. & Mch. Co., and *for mail*, 1160 W. High St., Lima, O.

WILLIAMS, Alan Gillespie (Junior, 1909), 112 Central Ave., South Oil City, Pa.
WOODS, Samuel Hamilton (Junior, 1907), Richardson & Boynton Co., 31 W. 31st St., and 73 W. 12th St., New York, N. Y.
YOUNG, William A. (1901; 1905; 1906), Ch. Draftsman, Morgan Engrg. Co., and *for mail*, 522 S. Arch St., Alliance, O.

NEW MEMBERS

ARNOTT, R. Fleming (Associate, 1909), Cons. Engr., 95-97 Liberty St., New York, N. Y.
BECK, Rudolph H. (Junior, 1909), United Iron Works, Seattle, Wash.
BONNEY, Herbert Marshall (Junior, 1909), Ch. Draftsman, L. F. Fales, and *for mail*, Walpole, Mass.
NELSON, Eric Hugo (1909), Ch. Draftsman, Geo. F. Blake Mfg. Co., Cambridge, and *for mail*, 61 Leverett Ave., Beachmont, Mass.

DEATHS

BLESSING, James H., February 21, 1910.
BRIDGE, James W., December 20, 1909.
CHURCHILL, William W., March 25, 1910.
HASKINS, Harry S., March 13, 1910.
REDWOOD, Iltyd I., April 5, 1910.
SIMS, Gardiner C., March 19, 1910.
SIRICH, J. Henry, Jr., January 22, 1910.

GAS POWER SECTION

CHANGES OF ADDRESS

ENGLISH, Harry K. (1909), Mem. Am. Soc. M. E.

FISKE, Geo. Wallace (Affiliate, 1909), 816 Huntoon St., Topeka, Kan.

HOPCROFT, Ernest Bigly (Affiliate, 1908), present address unknown.

LATHROP, Jay Cowden (Affiliate, 1908), Supt. of Constr., Peoples Ry. Co., Dayton, O.

LUMMIS, Charles W. (1908), Mem. Am. Soc. M. E.

MANGELSDORFF, Max F. (Affiliate, 1910), 115 Nassau St., New York, N. Y., and *for mail*, 212 Fifth St., Union Hill, N. J.

QUINN, Stephen (Affiliate, 1909), Ch. Engr., Iola Portland Cement Co., and *for mail* Clinker Club, 16 N. Buckeye St., Iola, Kan.

ROTH, Charles (Affiliate, 1909), Mech. Engr., L. A. Becker Co., Chicago, and *for mail*, 220 Marion St., Oak Park, Ill.

THOMPSON, Wm. K. (Affiliate, 1909), 2619 Orchard Ave., Los Angeles, Cal.

NEW MEMBERS

JAMES, Frederick Conway (Affiliate, 1910), Cons. Engr., Mangold Bros., Port Elizabeth, and *for mail*, Steytlerville, Cape Colony, South Africa.

STUDENT BRANCHES

CHANGES OF ADDRESS

BINNS, G. W. (Student, 1910), 2358 Ohio Ave., Cincinnati, O.
FRAMBACH, F. S. (Student, 1910), Hartley Hall, Columbia Univ., New York, N. Y.
HAINES, P. G. (Student, 1910), 3153 Willis Ave., Cincinnati, O.
HENWOOD, P. E. (Student, 1909), 4620 Calumet Ave., Chicago, Ill.
HOLLENBERGER, Theo. J. (Student, 1909), 4433 N. Paulina St., Chicago, Ill.
KARMAZIN, John (Student, 1910), 1001 S. 5th St., Champaign, Ill.
LURIE, A. N. (Student, 1909), 1871 S. Kedzie Ave., Chicago, Ill.
MARSH, Karl H. (Student, 1910), with Arthur G. McKee, Engr., Rockefeller Bldg., Cleveland, O.
MURDUCK, R. K. (Student, 1910), 705 W. Hill St., Champaign, Ill.
YOAKUM, F. E., Jr. (Student, 1909), 201 Dryden Rd., Ithaca, N. Y.

NEW MEMBERS

BROOKLYN POLYTECHNIC INSTITUTE

BREVOORT, C. (Student, 1910), 69 Lincoln Rd., Flatbush, N. Y.

COLUMBIA UNIVERSITY

DAVIS, F. R. (Student, 1910), 220 W. 107th St., New York, N. Y.
DEMOREST, W. J. (Student, 1910), 173 W. 93d St., New York, N. Y.
SWALLOW, Howard (Student, 1910), 605 W. 184th St., New York, N. Y.
VAUGHAN, L. L. (Student, 1910), Hartley Hall, Columbia Univ., New York, N. Y.
VON MAHLENBORG, C. A. (Student, 1910), 127 W. 111th St., New York, N. Y.

MASSACHUSETTS INSTITUTE OF TECHNOLOGY

ESTES, G. H. (Student, 1910), 31 Newbury St., Boston, Mass.
MACKENZIE, Morril (Student, 1910), 68 Barnum St., Taunton, Mass.
ROBB, Charles A. (Student, 1910), 12 Oxford St., Cambridge, Mass.
RUSSELL, Foster (Student, 1910), 80 St. Botolph St., Boston, Mass.
STONE, R. T. (Student, 1910), Tech. Chambers, Boston, Mass.

UNIVERSITY OF CINCINNATI

COLBURN, B. V. (Student, 1910), 2343 Stratford Ave., Cincinnati, O.
HUMPHREYS, H. B. (Student, 1910), 236 Ofallon Ave., Bellevue, Ky.
LYTLE, C. W. (Student, 1910), 2345 Stratford Ave., Cincinnati, O.

UNIVERSITY OF ILLINOIS

CARLSON, C. A. (Student. 1910), 1015 W. Illinois St., Urbana, Ill.
JACOBSEN, C. H. (Student, 1910), 906 W. Illinois St., Urbana, Ill.
LINDSTROM, A. W. (Student, 1910), 307 E. Daniel St., Champaign, Ill.
PAUL, Harry (Student, 1910), 209 E. Green St., Champaign, Ill.

COMING MEETINGS

MAY–JUNE

Advance notices of annual and semi-annual meetings of engineering societies are regularly published under this heading and secretaries or members of societies whose meetings are of interest to engineers are invited to send such notices for publication. They should be in the editor's hands by the 18th of the month preceding the meeting. When the titles of papers read at monthly meetings are furnished they will also be published.

AIR BRAKE ASSOCIATION

May 10–13, Dennisson Hotel, Indianapolis, Ind. Subjects for discussion, and chairmen: Air Brake Instruction, Examination and Rating, Thos. Clegg; Air Pump Piping, Fittings and Connections, George W. Kiehm; Best Arrangement of Air Pump and Main Reservoir Capacity for 100-car Train Service, P. J. Langan; Brake Cylinders and Connections to Cylinder Leakage, W. P. Garabrant; Inspection and Cleaning of Triple Valves and Brake Cylinders, C. P. McGinnis; Developments in Air Brakes, W. V. Turner; New York Brake Equipment, T. F. Lyons; Westinghouse Equipment, S. G. Down; Recommended Practice, S. G. Down. Secy., F. M. Nellis, 53 State St., Boston, Mass.

AMERICAN ASSOCIATION ELECTRIC MOTOR MANUFACTURERS

May 18, Newport News, Va. Secy., Frank H. Couch, Hampton.

AMERICAN CIVIC ALLIANCE

May 18, 29 W. 39th St., New York. Secy., Henry Frank.

AMERICAN EXPOSITION IN BERLIN

June 1-Aug. 31. American Manager, Max Vieweger, 50 Church St., New York.

AMERICAN FOUNDRYMEN'S ASSOCIATION
MANUFACTURERS SUPPLY ASSOCIATION

June 6–10, Detroit, Mich. Secy. of general committee, A. Preston Henry, Standard Pattern Works.

AMERICAN ELECTROCHEMICAL SOCIETY

May 4–7, Spring Meeting, Fort Pitt Hotel, Pittsburg, Pa. Among the papers will be: Furnace Conductors for Heavy Alternating Currents, K. C. Randall; A New Electric Steel Furnace, A. L. Queneau; Gases in Steel, P. L. T. Heroult; Cheap Power in the Pittsburg District, F. Crabtree; Induction Furnace Progress, T. Rowlands; Ductile Tungsten and Molybdenum, C. G. Fink; A New Process for the Treating of Cobalt-nickel Ores, C. C. Cito; A New Radiation Pyrometer, C. E. Foster, Mem. Am. Soc. M.E.; The Separation of Oil from Condenser Water by Electrolysis, H. M. Goodwin; A New Method for the Electrolytic Winning and Refining of Metals, E. M. Chance; Refining of Tin Dross in an Electric Furnace, R. S. Wile; The Effect of Moisture and of Solutions upon the Electrical

Conductivity of Soils, R. E. O. Davis; The Present Status of the Electro-Chemical Industries, J. W. Richards; Pittsburg as an Electrochemical Center, John A. Brashear, Mem.Am.Soc.M.E.; The Conservation and Utilization of Natural Sources of Power, J. H. Finney.

AMERICAN INSTITUTE OF CHEMICAL ENGINEERS

June 22-24, summer meeting, Niagara Falls, N. Y. Secy., J. C. Olsen, Polytechnic Inst., Brooklyn.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS

May 5-7, Auditorium of Home Telephone Co., 333 Grant Ave., San Francisco, Cal. Papers: Emergency Generating Stations for Service in Connection with Hydroelectric Transmission Plants under Pacific Coast Conditions, A. M. Hunt, Mem. Am. Soc. M. E.; Hydroelectric Power as Applied to Irrigation, J. C. Hays; The Developed High-Tension Net-Work of a General Power System, Paul M. Downing; Parallel Operation of Three-Phase Generators with their Neutrals Interconnected, G. I. Rhodes; Observations of Harmonics in Current and Potential Wave Shapes of Transformers, John J. Frank; Transmission Line Crossings of Railroad Right-of-way, A. H. Babcock. May 17, Annual Meeting; 33 W. 39th St., New York. May 27, special railway meeting, 33 W. 39th St. Papers: Application of Porcelain to Strain Insulators, W. H. Kempton; Electric Railway Overhead Construction, W. N. Smith. June 27-28, Annual Convention, Waumbeck Hotel, Jefferson, N. H. Secy., R. W. Pope, 33 W. 39th St.

AMERICAN PORTLAND CEMENT MANUFACTURERS

June, Kansas City, Kan. Secy., P. H. Wilson, Land Title Bldg., Philadelphia, Pa.

AMERICAN RAILWAY ACCOUNTING OFFICERS

June 29, Colorado Springs, Colo. Secy., C. G. Phillips, 143 Dearborn St., Chicago.

AMERICAN RAILWAY ASSOCIATION

May 18, 29 W. 39th St. New York, 11 a.m. Secy, W. F. Allen, 24 Park Pl.

AMERICAN RAILWAY INDUSTRIAL ASSOCIATION

May 10, Memphis, Tenn. Secy., Guy L. Stewart, S. W. Ry., St. Louis, Mo.

AMERICAN RAILWAY MASTER MECHANICS ASSOCIATION

June 20-22, Atlantic City, N. J. Secy., J. W. Taylor, 390 Old Colony Bldg., Chicago.

AMERICAN SOCIETY OF CIVIL ENGINEERS

May 4, 18, 220 W. 57th St., New York. Papers, May 4: Water Supply of El Paso & Southwestern Railway from Carrizozo to Santa Rosa, New Mexico, J. L. Campbell; New York Tunnel Extension of Pennsylvania Railroad: Site of the Terminal Station, G. C. Clarke. Secy., C. W. Hunt.

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

May 14, St. Louis, with coöperation of Engineers Club of St. Louis. May 31-June 3, Spring Meeting, Atlantic City, N. J. July 26-29, meeting with Institution of Mechanical Engineers, in Birmingham and London, England. Secy., Calvin W. Rice, 29 W. 39th St., New York.

AMERICAN SOCIETY FOR TESTING MATERIALS

June 28-July 2, annual meeting, Atlantic City, N. J. Secy., Edgar Marburg, University of Pennsylvania, Philadelphia.

ASSOCIATION OF CAR-LIGHTING ENGINEERS

June 7, 8, semi-annual convention, Buffalo, N. Y. Secy., Geo. B. Colegrave, care of Central Railway, Chicago.

BROOKLYN ENGINEERS CLUB

May 5, clubhouse, 117 Remsen St., 8 p.m. Paper: Steel Centering as Applied to the Catskill Aqueduct, V. R. Whitehall. May 14, visit to Pa. R. R. Sta., New York. Secy., Joseph Strachan.

CANADIAN GAS ASSOCIATION

June 9-11, annual convention, Alexander Rink, Hamilton, Ont. Secy., A. W. Moore, Woodstock, Ont.

CENTRAL RAILWAY CLUB

May 13, Buffalo, N. Y. Secy., H. D. Vought, 95 Liberty St., New York.

CLEVELAND ENGINEERING SOCIETY

June 14, annual meeting, 714 Caxton Bldg. Secy., J. C. Beardsley.

ENGINEERS' CLUB OF BALTIMORE

June 4, annual meeting. Secy., R. K. Compton, City Hall.

ENGINEERS SOCIETY OF MILWAUKEE

June 8, annual meeting, Builders Club. Secy., W. F. Martin, 456 Broadway.

ENGINEERS SOCIETY OF PENNSYLVANIA

June 7, annual meeting, Gilbert Bldg., Harrisburg. Secy., E. R. Dasher, P. O. Box 704.

ENGINEERS' CLUB OF ST. LOUIS

May 14, coöperating with Am. Soc. M. E. Secy., A. S. Landsdorf, 3817 Olive St.

FOUNDRY AND MANUFACTURERS' SUPPLY ASSOCIATION

June 6-10, Detroit, Mich. Secy., C. E. Hoyt, Lewis Institute, Chicago.

FREIGHT CLAIM ASSOCIATION

June 15, Los Angeles, Cal. Secy., W. P. Taylor, Richmond, Va.

INTERNATIONAL CONGRESS OF INVENTORS

June 13-18, Rochester, N. Y.

INTERNATIONAL CONGRESS OF MINING, METALLURGY, APPLIED MECHANICS AND PRACTICAL GEOLOGY

Last week in June, Düsseldorf, Prussia. Secy., Dr. E. Schrödter, Jacobstrasse 315.

INTERNATIONAL MASTER BOILER MAKERS' ASSOCIATION

May 24-26, New Clifton Hotel, Niagara Falls, Ont. Subjects for discussion and chairmen: Standardizing of Blue Prints, W. H. Laughridge; Application and Care of Flues, D. A. Lucas; Flexible Staybolts, C. J. Murray; Steel vs. Iron Tubes, M. O'Connor; Flue Holes in Back Flue Sheet, J. A. Dearnberger; Standardizing of Shop Tools, J. T. Goodwin; Standardizing of Pipe Flanges and Templates for Drilling, Jas. Crombie; Radical Departures in Boilers and Fire Boxes, B. F. Sarver; Fire Box Holes, W. H. Laughridge; Best Method of Staying Front Portion of Crown Sheet on Radical Top Boilers to prevent Cracking of Flue Sheet in Top Flange, H. J. Raps; Oxy-Acetylene Welding, M.O'Connor. Secy., H. D. Vought, 95 Liberty St., New York.

INTERNATIONAL RAILWAY FUEL ASSOCIATION

May 23-26, Chicago. Secy., D. B. Sebastian, 327 LaSalle St. Sta.

INTERNATIONAL RAILWAY GENERAL FOREMEN'S ASSOCIATION
May 3–7, Cincinnati, O. Secy., E. C. Cooke, Royal Ins. Bldg., Chicago.

IOWA DISTRICT GAS ASSOCIATION
June 15–17, annual meeting, Sioux City. Secy., G. I. Vincent, Des Moines.

LOUISIANA ENGINEERING SOCIETY
May 9, June 13, 321 Hibernia Bldg., New Orleans. Papers: The Manufacture of Sugar, W. H. P. Creighton; Coal in Relation to Boiler Economy, Wm. von Phul. Secy., L. C. Datz

MASTER CAR BUILDERS ASSOCIATION
June 15–17, Atlantic City, N. J. Secy., J. W. Taylor, 390 Old Colony Bldg., Chicago.

MUNICIPAL ENGINEERS OF THE CITY OF NEW YORK
May 27, inspection trip to Albany and Schenectady. Secy., C. D. Pollock, 29 W. 39th St., New York.

NATIONAL ASSOCIATION OF MANUFACTURERS
May 16–18, New York. Secy., George S. Boudinot, 170 Broadway.

NATIONAL DISTRICT HEATING ASSOCIATION
June 1–3, annual meeting, Toledo, O. Secy., D. C. Gaskill, Greenville, O.

NATIONAL ELECTRIC LIGHT ASSOCIATION
May 23–28, St. Louis, Mo., Secy., Frank H. Tate, Dayton, O.

NATIONAL ELECTRIC TRADES ASSOCIATION
June, San Francisco, Cal. Secy., F. B. Vose, 1343 Marquette Bldg., Chicago

NATIONAL FIRE PROTECTION ASSOCIATION
May 17–19, annual meeting, Hotel LaSalle, Chicago. Address by I. K. Pond, Pres. Am. Inst. Architects, etc. Secy., F. H. Wentworth, 87 Milk St., Boston.

NATIONAL GAS AND GASOLENE ENGINE TRADES ASSOCIATION
June 13–16, Semi-annual convention, Hotel Sinton, Cincinnati, O. Secy., Albert Stritmatter.

NATIONAL MACHINE TOOL-BUILDERS ASSOCIATION
May 24, 25, Spring Convention, Hotel Seneca, Rochester, N. Y. Secy., C. E. Hildreth, Worcester, Mass.

NATURAL GAS ASSOCIATION OF AMERICA
May 17–19. Oklahoma City, Okla. Secy., M. W. Walsh, 110 N. Broadway.

NEW ENGLAND WATERWORKS ASSOCIATION
June, Providence, R.I. September 14–16, annual convention, Rochester, N. Y. Secy., Willard Kent, Narragansett Pier, R. I.

NEW YORK RAILROAD CLUB
May 20, 29 W. 39th St., 8.15 p.m. Secy., H. D. Vought, 95 Liberty St.

OHIO SOCIETY OF ENGINEERS
May 19, 20, Cincinnati. Secy., F. E. Sanborn, State University, Columbus.

PROVIDENCE ASSOCIATION OF MECHANICAL ENGINEERS
May 24, West Hall, R. I. School of Design, 8 p. m. Paper: Modern Machine Tools. Second week in May, Spring visitation, The Norton Co., Worcester, Mass., June 28, annual meeting. Secy., T. M. Phetteplace, Mem. Am.Soc.M.E., 48 Snow St.

RAILWAY SIGNAL ASSOCIATION
June 14, 29 W. 39th St., New York, 9.30 a.m. Secy., C. C. Rosenberg, Bethlehem, Pa.

RENSSELAER SOCIETY OF ENGINEERS

June, annual meeting, Rensselaer Polytechnic Inst., 257 Broadway, Troy, N. Y. Secy., R. S. Furber.

ST. LOUIS RAILWAY CLUB

May 13. Secy., B. W. Frauenthal, Union Station.

SOCIETY FOR PROMOTION OF ENGINEERING EDUCATION

June 23–25, Madison, Wis. Papers on Technical Education Abroad, Inspection Trips for Technical Students. Efficiency in Technical Education. Secy., H. H. Norris, Cornell University, Ithaca. N. Y.

STEVENS ENGINEERING SOCIETY

May 3, 10, Hoboken, N. J. Papers by C. F. Kroeh and J. A. Brashear, Hon. Mem.Am.Soc.M.E.

TELEPHONE SOCIETY OF NEW YORK

June 21, annual meeting, 29 W. 39th St. Secy., T. H. Woolhouse.

TRANSPORTATION AND CAR ACCOUNTING OFFICERS

June 28. Secy., G. P. Conard, 24 Park Pl., New York.

WESTERN CANADA RAILWAY CLUB

May 9, annual meeting, Royal Arms Hotel, Winnipeg, Manitoba. Secy., W. H. Rosevear, P. O. Box 1707.

MEETINGS IN THE ENGINEERING SOCIETIES BUILDING

Date	Society	Secretary	Time
May			p. m.
4	Wireless Institute	S. L. Williams	7.30
5	Blue Room Engineering Society	W. D. Sprague	8.00
6	Western Union Electrical Society	H. C. Northen	7.00
7	Amer. Soc. Hungarian Engrs. and Archts.	Z. deNemeth	8.30
12	Illuminating Engineering Society	P. S. Millar	8.00
13	Western Union Electrical Society	H. C. Northen	7.00
17	American Institute of Electrical Engineers	R. W. Pope	8.00
17	New York Telephone Society	T. H. Lawrence	8.00
			a. m.
18	American Railway Association	W. F. Allen	11.00
			p. m.
18	American Civic Alliance	Henry Frank	2.30 8.00
20	New York Railroad Club	H. D. Vought	8.15
20	Western Union Electrical Society	H. C. Northen	7.00
25	Municipal Engineers of the City of New York	C. D. Pollock	8.15
27	American Institute of Electrical Engineers	R. W. Pope	8.00
27	Western Union Electrical Society	H. C. Northen	7.00
June			
1	Wireless Institute	S. L. Williams	7.30
2	Blue Room Engineering Society	W. D. Sprague	8.00
3	Western Union Electrical Society	H. C. Northen	7.00
4	Amer. Soc. Hun. Engrs. and Archts.	Z. deNemeth	8.30
9	Illuminating Engineering Society	P. S. Millar	8.15
10	Western Union Electrical Society	H. C. Northen	7.00

Date	Society	Secretary	Time
June			a.m.
14	Railway Signal Association	C. C. Rosenberg	9.30
			p. m.
17	Western Union Electrical Society	H. C. Northen	7.00
21	New York Telephone Society	T. H. Lawrence	8.00
24	Western Union Electrical Society	H. C. Northen	7.00

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Terms expire at Annual Meeting of 1910

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W. J. SANDO ..... Milwaukee, Wis.

Terms expire at Annual Meeting of 1911

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NOTE—Numbers in parentheses indicate number of years the member is yet to serve.

SPECIAL COMMITTEES

1910

On a Standard Tonnage Basis for Refrigeration

D. S. JACOBUS — G. T. VOORHEES
A. P. TRAUTWEIN — PHILIP DE C. BALL
E. F. MILLER

On Society History

JOHN E. SWEET — H. H SUPLEE
CHAS. WALLACE HUNT

On Constitution and By-Laws

CHAS. WALLACE HUNT, *Chairman* — F. R. HUTTON
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JESSE M. SMITH

On Conservation of Natural Resources

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On International Standard for Pipe Threads

E. M. HERR, *Chairman* — GEO. M. BOND
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On Standards for Involute Gears

WILFRED LEWIS, *Chairman* — E. R. FELLOWS
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GAETANO LANZA

On Power Tests

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GEORGE H. BARRUS — CHARLES E. LUCKE — ALBERT C. WOOD

On Student Branches

F. R. HUTTON, HONORARY SECRETARY

On Meetings of the Society in Boston

IRA N. HOLLIS, *Chairman* — I. E. MOULTROP, *Secretary*
EDWARD F. MILLER — J. H. LIBBEY
CHARLES T. MAIN

On Meetings of the Society in St. Louis

WM. H. BRYAN, *Chairman* — ERNEST L. OHLE, *Secretary*
M. L. HOLMAN

SOCIETY REPRESENTATIVES

1910

On John Fritz Medal

AMBROSE SWASEY (1)
F. R. HUTTON (2)
CHAS. WALLACE HUNT (3)
HENRY R. TOWNE (4)

On Board of Trustees United Engineering Societies Building

F. R. HUTTON (1)
FRED J. MILLER (2)
JESSE M. SMITH (3)

On Library Conference Committee

J. W. LIEB, JR., CHAIRMAN OF THE LIBRARY COMMITTEE, AM. SOC. M. E.

On National Fire Protection Association

JOHN R. FREEMAN
IRA H. WOOLSON

On Joint Committee on Engineering Education

ALEX. C. HUMPHREYS
F. W. TAYLOR

On Government Advisory Board on Fuels and Structural Materials

GEO. H. BARRUS
P. W. GATES
W. F. M. GOSS

On Advisory Board National Conservation Commission

GEO. F. SWAIN
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ALEX. C. HUMPHREYS
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1909-1910

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NOTE—Numbers in parentheses indicate number of years the member is yet to serve.

OFFICERS OF STUDENT BRANCHES

INSTITUTION	BRANCH AUTHORIZED BY COUNCIL	HONORARY CHAIRMAN	PRESIDENT	CORRESPONDING SECRETARY
	1908			
Stevens Inst. of Tech., Hoboken, N. J.	December 4	Alex. C. Humphreys	H. H. Haynes	R. H. Upson
Cornell University, Ithaca, N. Y.	December 4	R. C. Carpenter	.....	F. E. Yoakum, Jr.
	1909			
Armour Inst. of Tech., Chicago, Ill.	March 9	G. F. Gebhardt	F. E. Wernick	W. E. Thomas
Leland Stanford, Jr. University, Palo Alto, Cal.	March 9	W. F. Durand	A. F. Meston	J. B. Bubb
Polytechnic Institute, Brooklyn, N. Y.	March 9	W. D. Ennis	J. S. Kerins	Percy Gianella
State Agri. College, Corvallis, Ore.	March 9	Thos. M. Gardner	C. L. Knopf	S. H. Graf
Purdue University, Lafayette, Ind.	March 9	L. V. Ludy	E. W. Templin	H. A. Houston
Univ. of Kansas, Lawrence, Kan.	March 9	P. F. Walker	C. E. Johnson	C. A. Swiggett
New York Univ., New York	November 9	C. E. Houghton	Harry Anderson	Andrew Hamilton
Univ. of Illinois, Urbana, Ill.	November 9	W. F. M. Goss	B. L. Keown	C. S. Huntington
Penna. State College, State College, Pa.	November 9	J. P. Jackson	G. B. Wharen	G. W. Jacobs
Columbia University, New York	November 9	Chas. E. Lucke	F. R. Davis	H. B. Jenkins
Mass. Inst. of Tech., Boston, Mass.	November 9	Gaetano Lanza	Morril Mackenzie	Foster Russell
Univ. of Cincinnati, Cincinnati, O.	November 9	J. T. Faig	W. H. Montgomery	P. G. Haines
Univ. of Wisconsin, Madison, Wis.	November 9	C. C. Thomas	John S. Langwill	Karl L. Kratz
Univ. of Missouri, Columbia, Mo.	December 7	H. Wade Hibbard	R. E. Dudley	F. T. Kennedy
Univ. of Nebraska, Lincoln, Neb.	December 7	C. R. Richards	M. E. Strieter	A. D. Stancliff
	1910			
Univ. of Maine, Orono, Me.	February 8	.....	H. N. Danforth	A. H. Blaisdell
Univ. of Arkansas, Fayetteville, Ark.	April 12	.....	.....	.....